Energy Systems course
Lecture notes

Hydraulic Turbines and Hydroelectric Power Plants

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Hydraulic Turbines and Hydroelectric Power Plants

Contents

1. Hydraulic turbines

   - Fundamental operating parameters
   - Classification
     - **Impulse turbines**
       - Pelton turbines
     - **Reaction turbines**
       - Radial flow: Francis turbines
       - Axial flow: propeller (fixed blades) or Kaplan (variable pitch blades) turbines
     - **Reversible pump-turbines**

2. Hydroelectric power plants

   - **Run-of-the-river**: small amounts of water storage -> little control of the flow through the plant
   - **Storage**: an artificial basin (created by a dam on a river course) allows to store water and thus control the flow through the plant on a daily or seasonal basis
   - **Pumped storage**: during off-peak hours water is pumped (by means of reversible pump-turbines or dedicated pumps) from a lower reservoir to an upper reservoir -> energy is thus stored for later production during peak hours

Hydraulic Turbines and Hydroelectric Power Plants

Gross and net head

Gross head is the difference between hydraulic heads in the upstream and downstream reservoirs:

$$H_g = H_u - H_d =$$

$$z_u - z_d + \frac{p_u - p_d}{\rho g} + \frac{c_u^2 - c_d^2}{2g}$$

Usually the only non negligible contribution comes from the geodetic head:

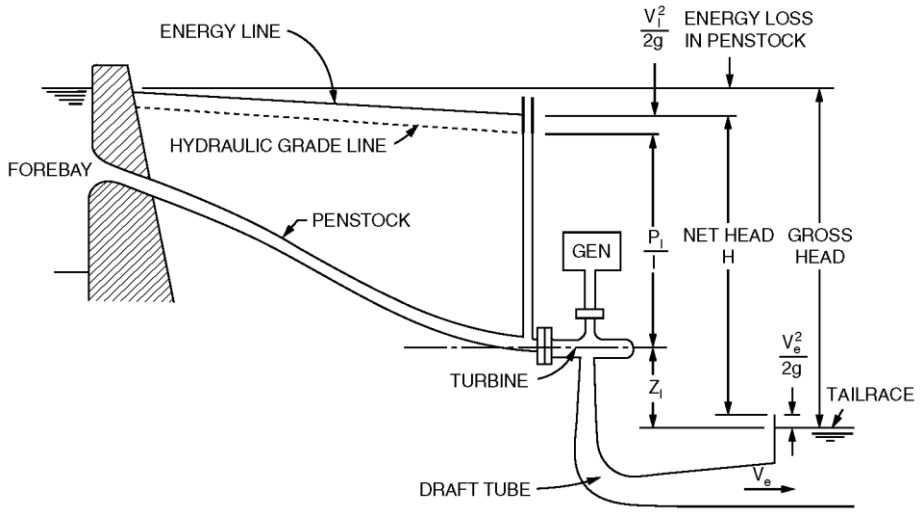
$$H_g = z_u - z_d$$

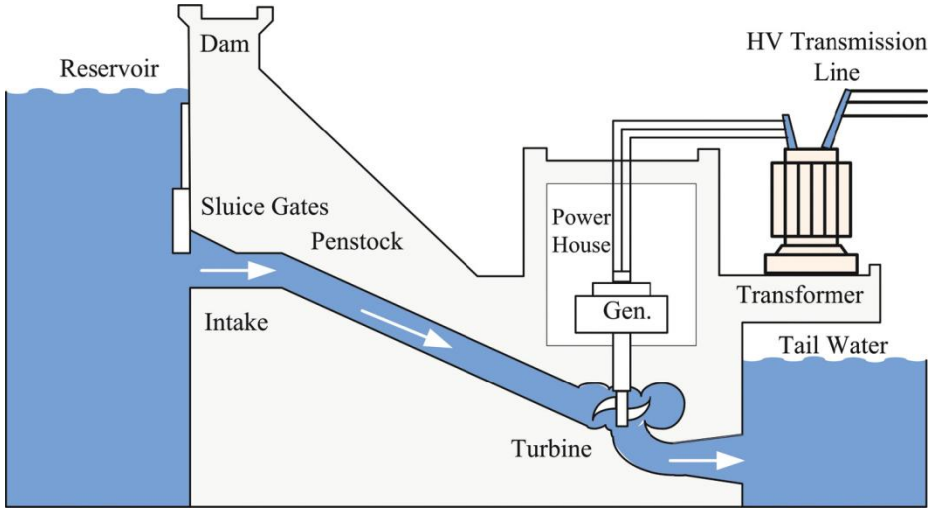
Net head is lower than gross head due to energy losses in the penstock:

$$H = H_g - Y$$

Penstock efficiency is the ratio of net and gross head:

$$\eta_p = \frac{H}{H_g} = 1 - \frac{Y}{H_g}$$





Hydraulic Turbines

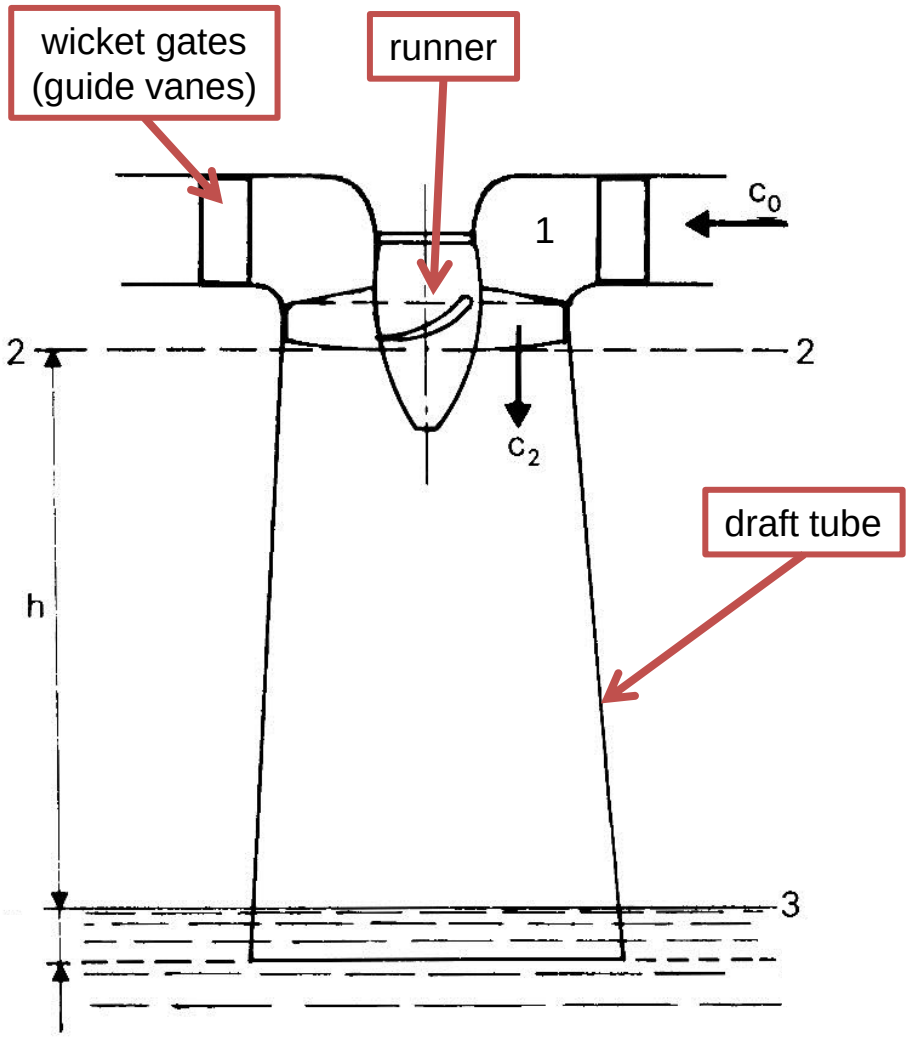
Constitutive elements of reaction turbines

The most important constitutive elements of reaction turbines are the following:

1. **wicket gates (or guide vanes)**
   vanes that guide water onto the runner, with appropriate velocity and direction

2. **runner**
   connected to the rotating shaft, it extracts energy from the water flow that interacts with its **blades**

3. **draft tube**
   if water's kinetic energy is still relatively high at the runner's exit, a draft tube is used to recover part of this kinetic energy



Hydraulic Turbines

Power and efficiencies

Hydraulic efficiency:

$$\eta_y = \frac{W}{gH} = 1 - \frac{\Sigma H_{l,t}}{H}$$

Turbine losses:

$$\Sigma H_{l,t} = H_{l,wg} + H_{l,r} + H_{l,dt} + \frac{c_3^2}{2g}$$

Volumetric efficiency:

$$\eta_v = \frac{Q_u}{Q}$$

Gross power output**:**

$$P_g = \rho Q_u W = \eta_v \, \eta_y \, \rho Q g H$$

Net power output:

$$P = P_g - P_m - P_{aux}$$

Generator efficiency (including mechanical and auxiliary losses):

$$\eta_g = \frac{P}{P_g} = 1 - \frac{P_m + P_{aux}}{P_g}$$

Net power output:

$$P = \eta_v \, \eta_y \, \eta_g \, \rho Q g H = \eta_t \rho Q g H$$

Overall turbine efficiency:

$$\eta_t = \eta_v \, \eta_y \, \eta_g$$

Overall plant efficiency:

$$\eta = \eta_t \, \eta_p = \frac{P}{\rho Q g H_g}$$

Hydraulic Turbines

Stage reaction

If the working fluid is incompressible, its enthalpy change in an adiabatic process depends on pressure change:

$$\Delta h = \Delta p/\rho$$

Energy conservation gives the work per unit mass:

$$W = \Delta h + g\Delta z + \frac{\Delta c^2}{2} = g\Delta H_p + \frac{\Delta c^2}{2}$$

Piezometric head H_p is the sum of **pressure head** ($p/\rho g$) and **elevation head** (z):

$$H_p = z + p/\rho g$$

Stage reaction in a hydraulic turbine is the ratio of piezometric head change in the runner and draft tube and the total piezometric head change:

$$R = \frac{\Delta H_{p,r}}{\Delta H_p}$$

For **ideal working conditions ($\eta_y = 1$)** total piezometric head change is equal to the work output (neglecting the difference between inlet and outlet kinetic energy) :

$$g\Delta H_p = gH = W$$

Energy conservation equation, applied between runner inlet (1) and draft tube outlet (3), yields:

$$z_1 + \frac{p_1}{\rho g} + \frac{c_1^2}{2g} = z_3 + \frac{p_3}{\rho g} + \frac{c_3^2}{2g} + \frac{W}{g} \cong z_3 + \frac{p_3}{\rho g} + \frac{W}{g}$$

Therefore:

$$g\Delta H_{p,r} = W - \frac{c_1^2}{2}$$

$$R = 1 - \frac{c_1^2}{2gH}$$

Water velocity at the runner inlet therefore depends on net head and stage reaction:

$$c_1 = \sqrt{2(1-R)gH}$$

Hydraulic Turbines

Specific speed

From dimensional analysis, it turns out that the turbine's most significant operating parameters:

- rotational speed n
- volumetric flow rate Q
- net head H

can be summed up in a single dimensionless parameter, which is invariant for geometrically similar turbines working under conditions of kinematic similarity. This parameter is the **specific speed**:

$$\boldsymbol{n_s = n\frac{Q^{1/2}}{H^{3/4}}}$$

The specific speed thus defined is not truly dimensionless, so its value may change if different units of measure or definitions are used.

For example, an alternative definition that is commonly used, which gives different numeric values even with the same units of measure, is the following, where power substitutes flow rate:

$$n_s' = n\frac{P^{1/2}}{H^{5/4}}$$

Hydraulic Turbines

Specific speed

Specific speed is usually calculated with rotational speed in [rpm], flow rate in [m^3/s], head in [m].

The truly dimensionless parameter, corresponding to the specific speed, is obtained substituting angular speed ω to rotation speed and available energy per unit mass gH to head:

$$\boldsymbol{k = \omega \frac{Q^{1/2}}{(gH)^{3/4}}}$$

The ratio between k and n_s is:

$$\frac{k}{n_s} = \frac{2\pi}{60 g^{3/4}} = 1{,}89 \cdot 10^{-2}$$

Hydraulic Turbines

Specific speed

The specific speed, being a dimensionless parameter, depends only on geometric and kinematic parameters.

$$n_s \propto \frac{u_1}{D_1} (l_1 D_1 c_1 \sin \alpha_1)^{\frac{1}{2}} H^{-\frac{3}{4}}$$

Hydraulic head is related to tip speed, water speed (Euler equation) and hydraulic efficiency:

$$H \propto \frac{u_1 c_{1u}}{\eta_y} = \frac{u_1 c_1 \cos \alpha_1}{\eta_y}$$

The specific speed thus becomes:

$$n_s \propto \left(\frac{l_1}{D_1}\right)^{\frac{1}{2}} \left(\frac{u_1}{c_1}\right)^{\frac{1}{4}} (\tan \alpha_1)^{\frac{1}{2}} (\cos \alpha_1)^{-\frac{1}{4}}$$

Making use of stage reaction:

$$W = 2(1-R)u_1^2 = u_1 c_{1u} = u_1 c_1 \cos \alpha_1 \Rightarrow \boldsymbol{\frac{u_1}{c_1} = \frac{\cos \alpha_1}{2(1-R)}}$$

the specific speed is finally:

$$\boldsymbol{n_s \propto \left(\frac{l_1}{D_1}\right)^{\frac{1}{2}} \left(\frac{1}{1-R}\right)^{\frac{1}{4}} (\tan \alpha_1)^{\frac{1}{2}}}$$

Hydraulic Turbines

Other quasi-dimensionless parameters

Other useful dimensionless parameters, which are used to describe the performance of a family of turbines, describe rotational speed and flow rate with reference to turbine size and hydraulic head:

$$\nu = n\frac{D}{\sqrt{H}}$$

$$q = \frac{Q}{D^2\sqrt{H}}$$

These parameters are useful to describe the behavior of geometrically similar turbines, and are related in an obvious way to the specific speed:

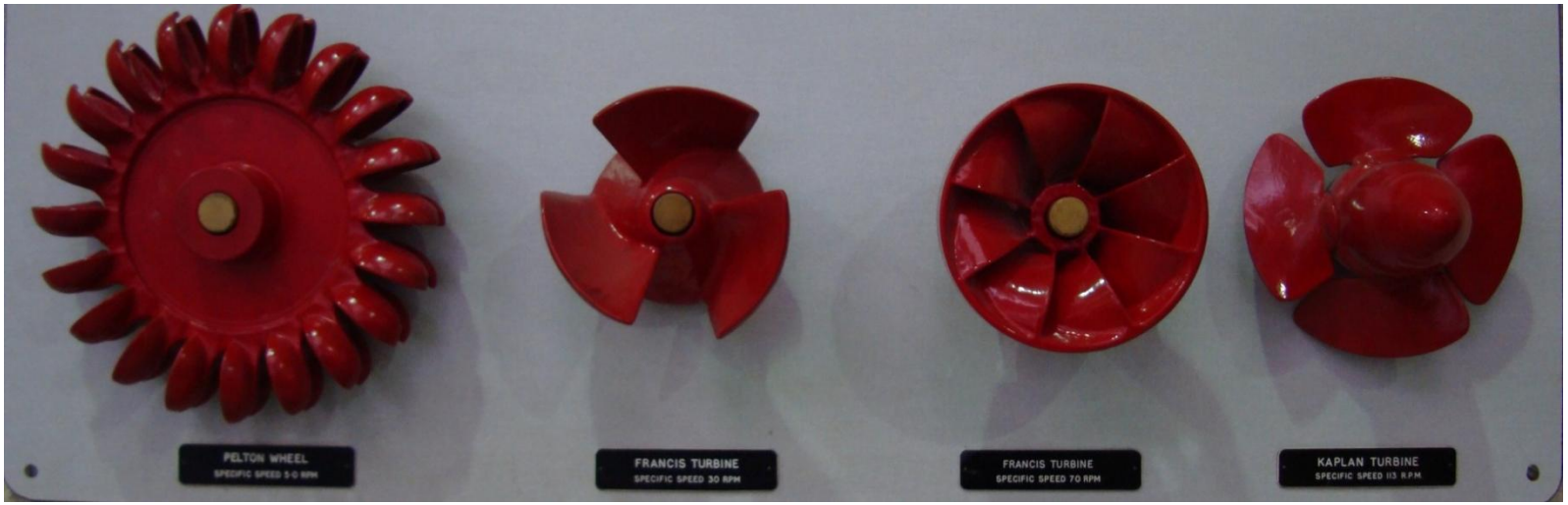
$$\nu q^{1/2} = n\frac{Q^{1/2}}{H^{3/4}} = n_s$$

Hydraulic Turbines

Classification

Hydraulic turbines :

- **Impulse turbines**: hydraulic head is converted to kinetic energy before water enters the runner.
  - **Pelton turbines**
- **Reaction turbines**: the runner is completely submerged and both pressure and velocity decrease from runner inlet to outlet.
  - **Francis turbines** (radial or mixed flow)
  - **Axial turbines** (axial flow): **Kaplan** (adjustable blade pitch), **propeller** (fixed blade pitch)



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Hydraulic Turbines

Classification

This table gives an overview of reference values of specific speed and stage reaction for different hydraulic turbines.

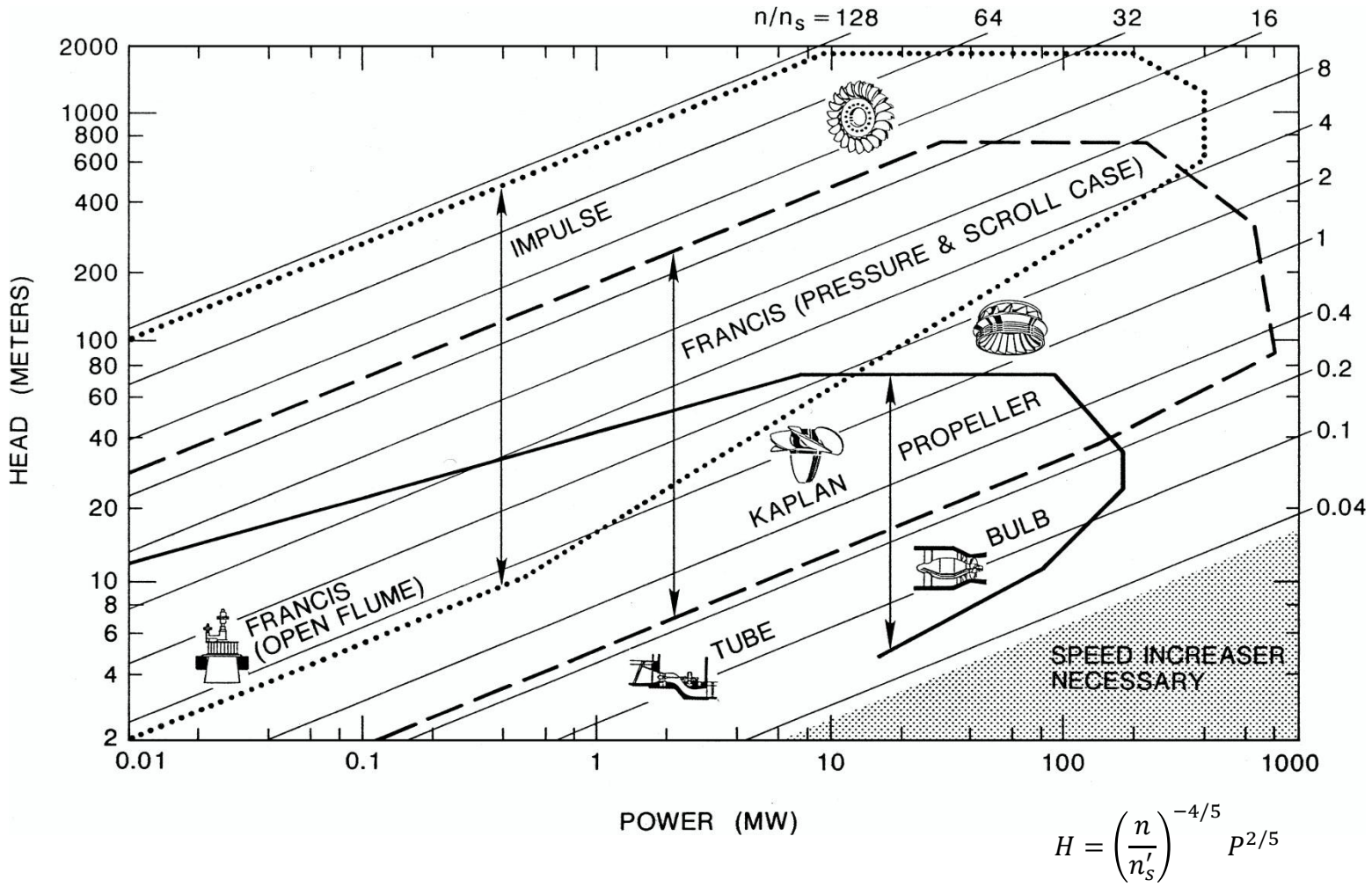
The specific speed increases as flow rate increases and hydraulic head decreases.

Therefore, turbines with high specific speed have also high values of stage reaction, because work exchanged between fluid and runner decreases if R increases.

	k	n_s	R
Pelton 1 jet	0,05 ÷ 0,2	5 ÷ 10	0
Pelton 2 jets	0,1 ÷ 0,3	7 ÷ 14	0
Pelton (>2 jets)	0,3 ÷ 0,4	14 ÷ 20	0
Francis (“slow”)	0,3 ÷ 0,6	15 ÷ 33	0,30
Francis (“medium”)	0,6 ÷ 1,0	33 ÷ 55	0,40
Francis (“fast”)	1,0 ÷ 1,6	55 ÷ 80	0,50
Francis (“ultrafast”)	1,6 ÷ 2,3	80 ÷ 120	0,60
Propeller, Kaplan	1,4 ÷ 5,7	75 ÷ 300	0,70

Hydraulic Turbines

Classification



$$H = \left(\frac{n}{n_s'}\right)^{-4/5} P^{2/5}$$

Specific speed expressed as $nP^{1/2}H^{-5/4}$

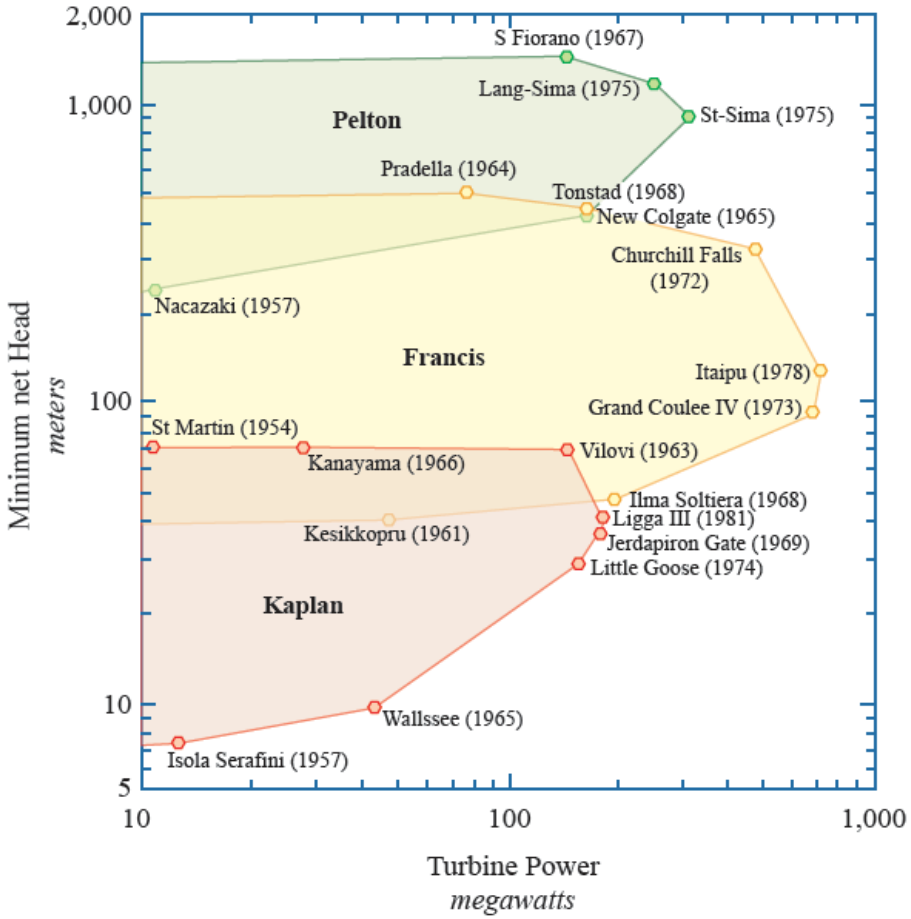
Source: John S. Gulliver, Roger E.A. Arndt, *Hydroelectric Power Stations, In: Encyclopedia of Physical Science and Technology (Third Edition)*, Academic Press, New York, 2003, Pages 489-504, ISBN 9780122274107, 10.1016/B0-12-227410-5/00321-5. (http://www.sciencedirect.com/science/article/pii/B0122274105003215)

Hydraulic Turbines

Classification

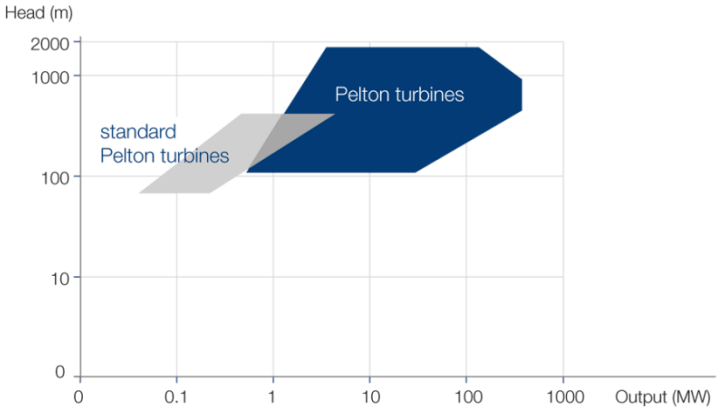
Reference values of working parameters and outputs for the main types of hydraulic turbines:

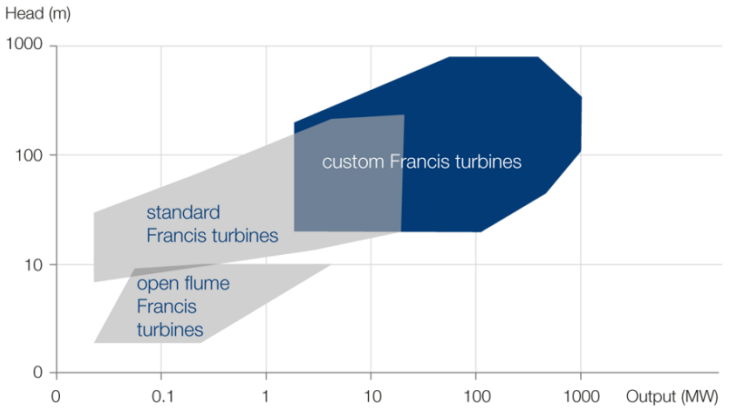
- Pelton:
  - flow rate ~ 0,5 ÷ 20 m^3/s
  - head ~ 300 ÷ 1500 m
  - net power up to ~ 200 MW
- Francis:
  - flow rate ~ 2 ÷ 800 m^3/s
  - head ~ 50 ÷ 400 m
  - net power up to ~ 800 MW
- Kaplan:
  - flow rate up to ~ 1000 m^3/s
  - head up to ~ 40 m
  - net power up to ~ 200 MW

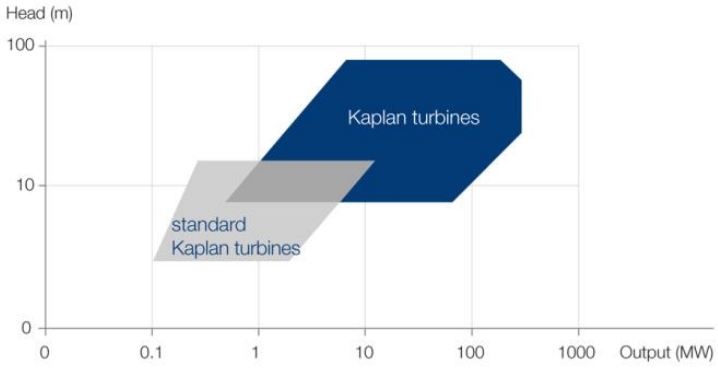


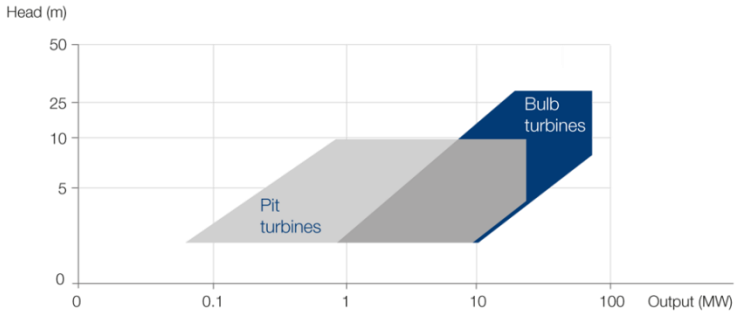
Hydraulic Turbines

Classification







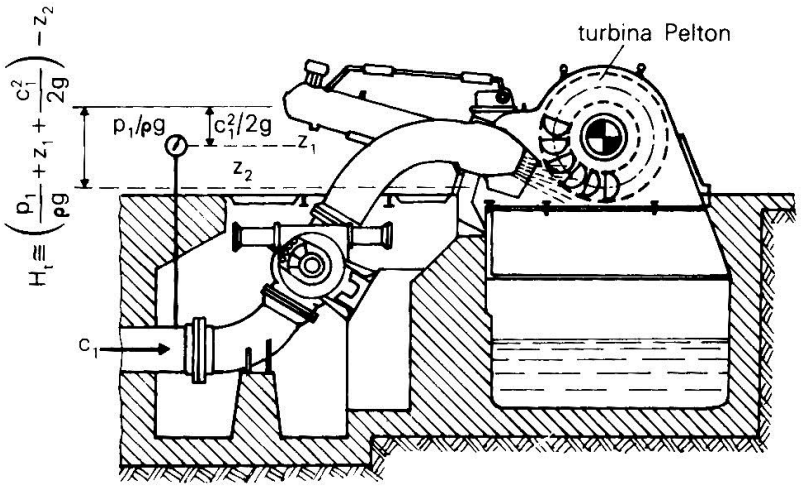


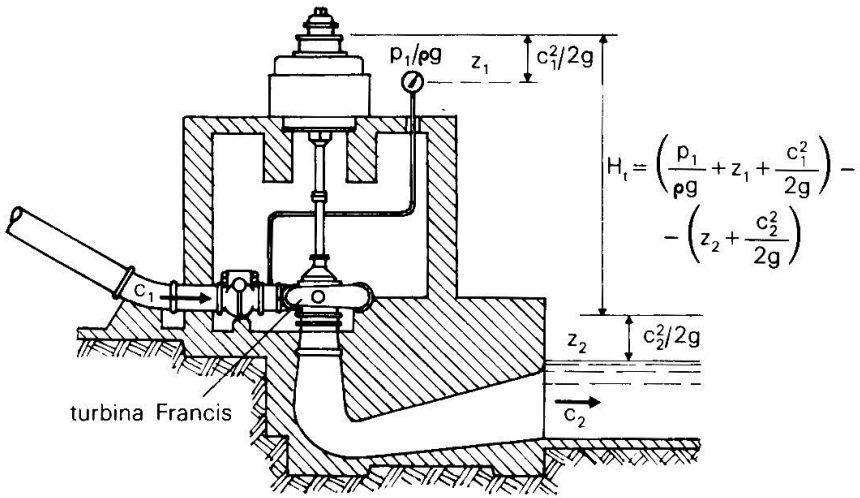
Source: Voith-Siemens

Hydraulic Turbines

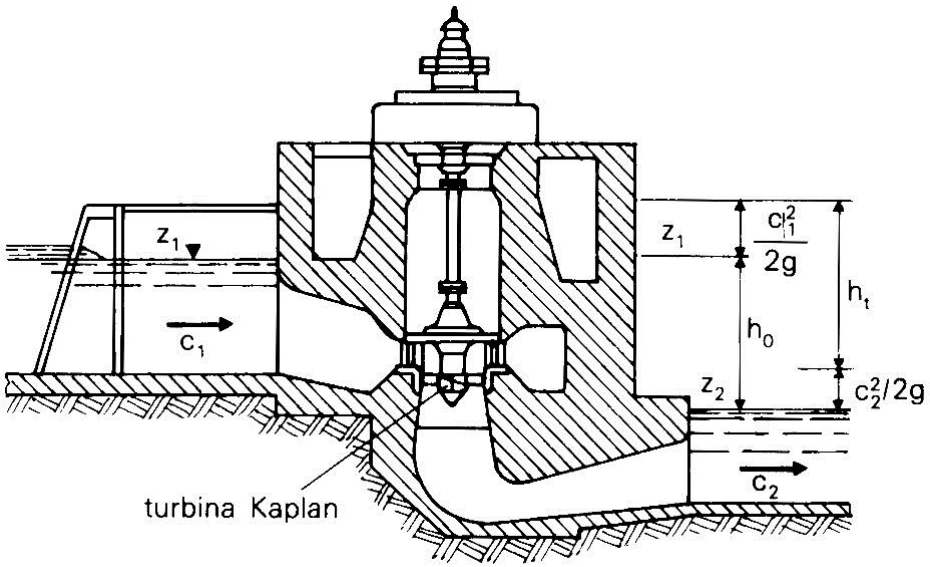
Classification

High head power plant





Medium head power plant



Low head power plant

Hydraulic Turbines

Draft tube

In case of high flow rates and relatively low hydraulic heads, it becomes impossible to decrease kinetic energy at sufficiently low values directly at the runner exit.

Therefore, a **draft tube** is necessary in order to recover as much kinetic energy as possible.

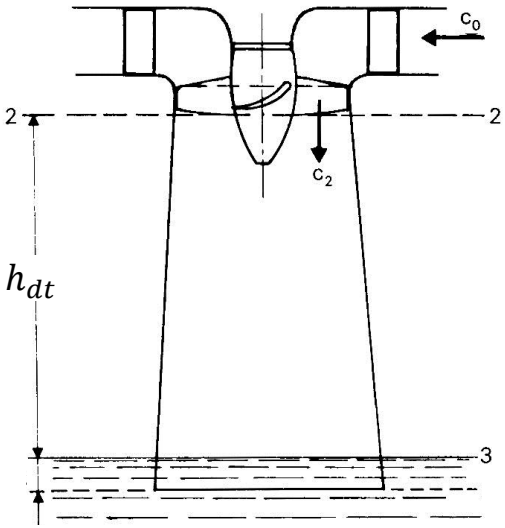
The induced depression at the runner exit must not determine the onset of **cavitation**.

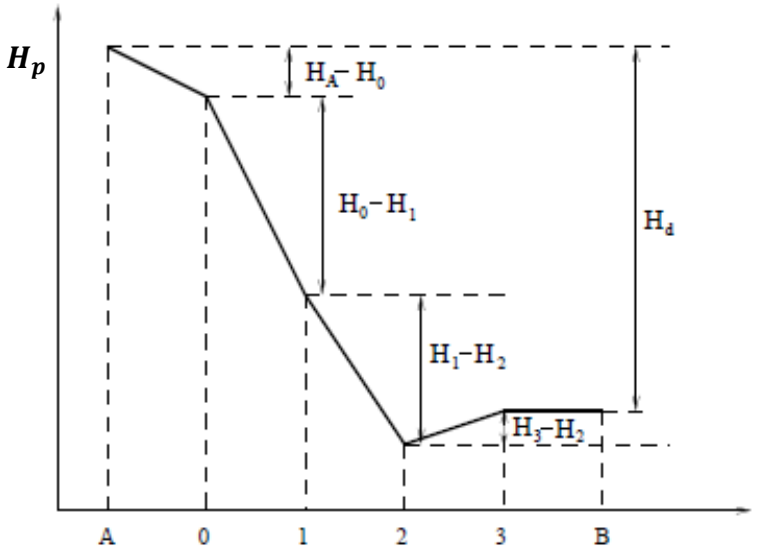
Energy conservation applied between runner exit and tailrace:

$$z_2 + \frac{p_2}{\rho g} + \frac{c_2^2}{2g} = z_3 + \frac{p_3}{\rho g} + \frac{c_3^2}{2g} + Y_{dt}$$

$$\cong z_3 + \frac{p_3}{\rho g} + Y_{dt}$$

setting $c_3 \cong 0$.





Piezometric head as water runs from upstream to downstream reservoir

Hydraulic Turbines

Cavitation and turbine setting

Minimum pressure values are found on blade suction side at the runner exit:

$$\frac{p_{min}}{\rho g} = \frac{p_2}{\rho g} - \Delta p$$

Since $p_3 = p_{atm}$, neglecting the partial pressure of dissolved air in water, in order to avoid cavitation the minimum pressure must be above vapor pressure ($p_{min} \geq p_v$), so the **maximum turbine elevation above tailrace $\boldsymbol{z = z_2 - z_3}$** (also called **turbine setting**) is given by:

$$z \leq \frac{p_{atm} - p_v}{\rho g} - \frac{\Delta p}{\rho g} - \frac{c_2^2}{2g} - Y_{dt}$$

A more convenient expression may be obtained if all terms that depend only on the turbine (and not on power plant characteristics) are grouped:

$$\Delta H_t = \frac{\Delta p}{\rho g} + \frac{c_2^2}{2g} + Y_{dt}$$

The final equation for draft tube maximum height can thus be written as follows:

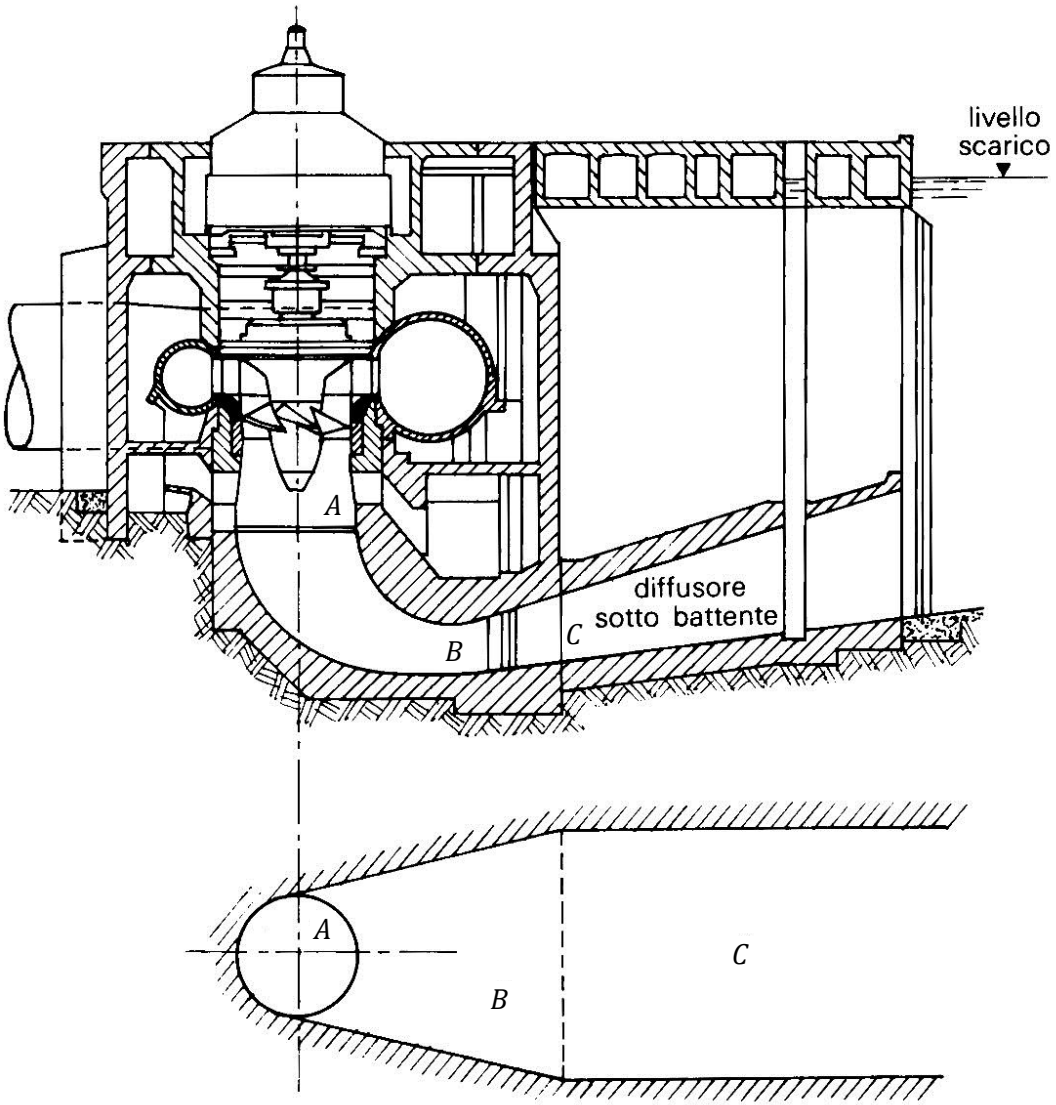
$$z \leq \frac{p_{atm} - p_v}{\rho g} - \Delta H_t$$

Hydraulic Turbines

Cavitation and draft tube

In many cases, in order to avoid cavitation it is necessary to place the runner exit below the tailrace level (under head).

In such situations the draft tube must have a curved geometry.



Hydraulic Turbines

Thoma cavitation coefficient

The inequality can be rearranged as follows:

$$(p_{atm} - p_v)/\rho g - z \geq \Delta H_t$$

Left hand side depends only on plant characteristics (the draft tube is considered part of the turbine): it is usually compared to net hydraulic head H by means of **Thoma cavitation coefficient σ**:

$$\boldsymbol{\sigma = \frac{(p_{atm} - p_v)/\rho g - z}{H}}$$

In order to avoid cavitation, Thoma coefficient must be higher than a critical threshold value σ_c that depends on ΔH_t:

$$\boldsymbol{\sigma_c = \Delta H_t / H}$$

Therefore:

$$\boldsymbol{\sigma \geq \sigma_c}$$

Typical values of σ for different specific speeds:

	Francis	Francis	Francis	Francis	Francis	Kaplan	Kaplan	Kaplan
n_s	20	40	60	80	100	100	150	200
σ_c	0,025	0,1	0,23	0,4	0,64	0,43	0,73	1,5

Source: R. L. Dougherty, J. B. Franzini, E. J. Finnemore, *Fluid Mechanics with Engineering Applications*, 8th ed., McGraw-Hill, New York (1985).

Hydraulic Turbines

Cavitation

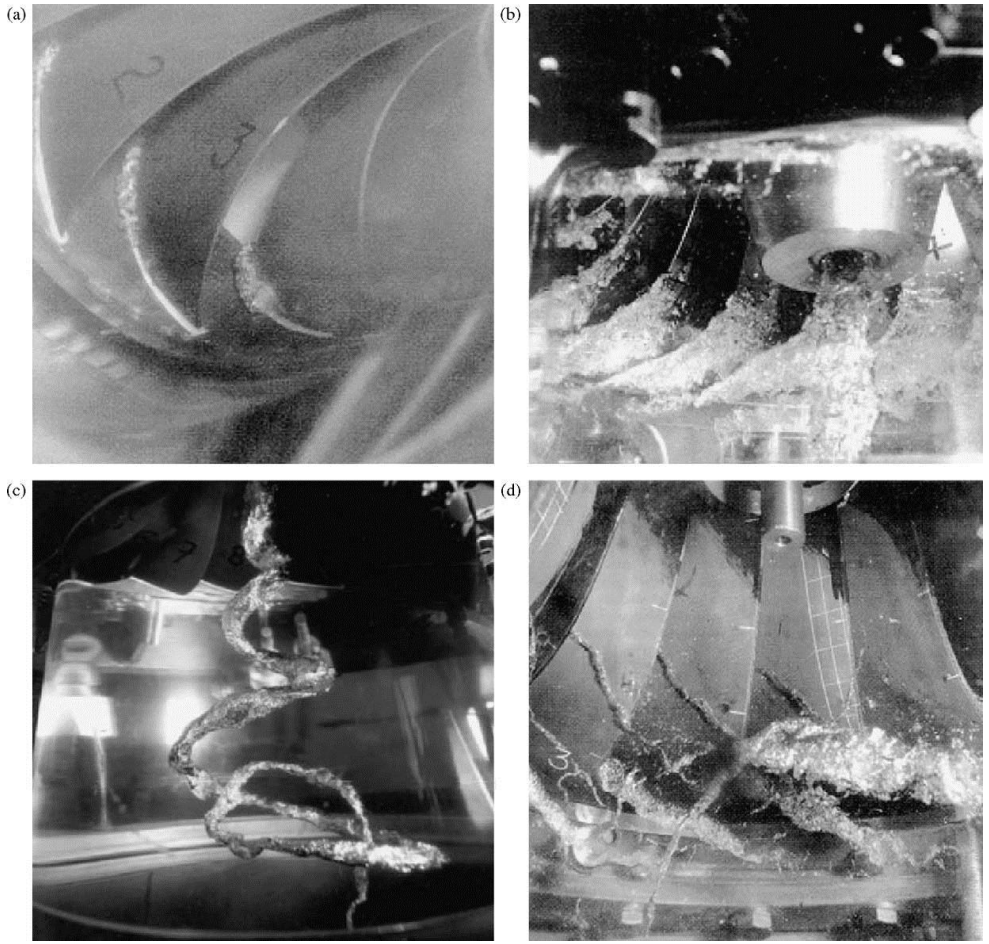
Main forms of cavitation on Francis turbines:

a) Leading edge cavitation
it takes the form of an attached cavity on the suction side of the runner blades due to higher than nominal heads

b) Travelling bubble cavitation
it takes the form of separated bubbles attached to the blade suction side near the mid-chord next to the trailing edge

c) Draft tube swirl
cavitation vortex-core flow that is formed just below the runner cone in the center of the draft tube

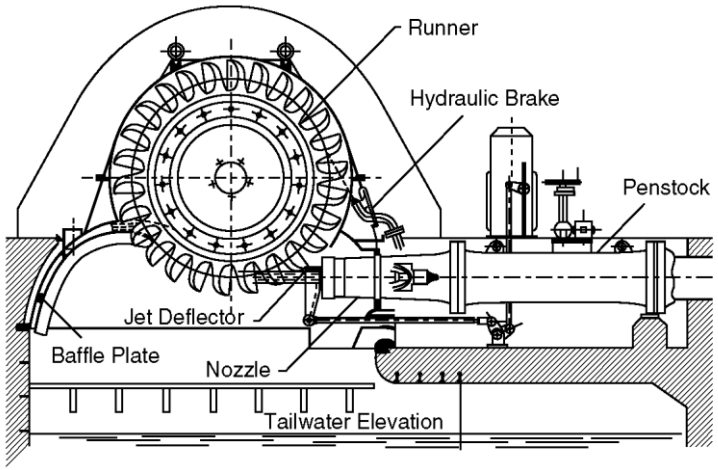
d) Inter-blade vortex cavitation
it is formed by secondary vortices located in the channels between blades that arise due to the flow separation provoked by the incidence variation from the hub to the band

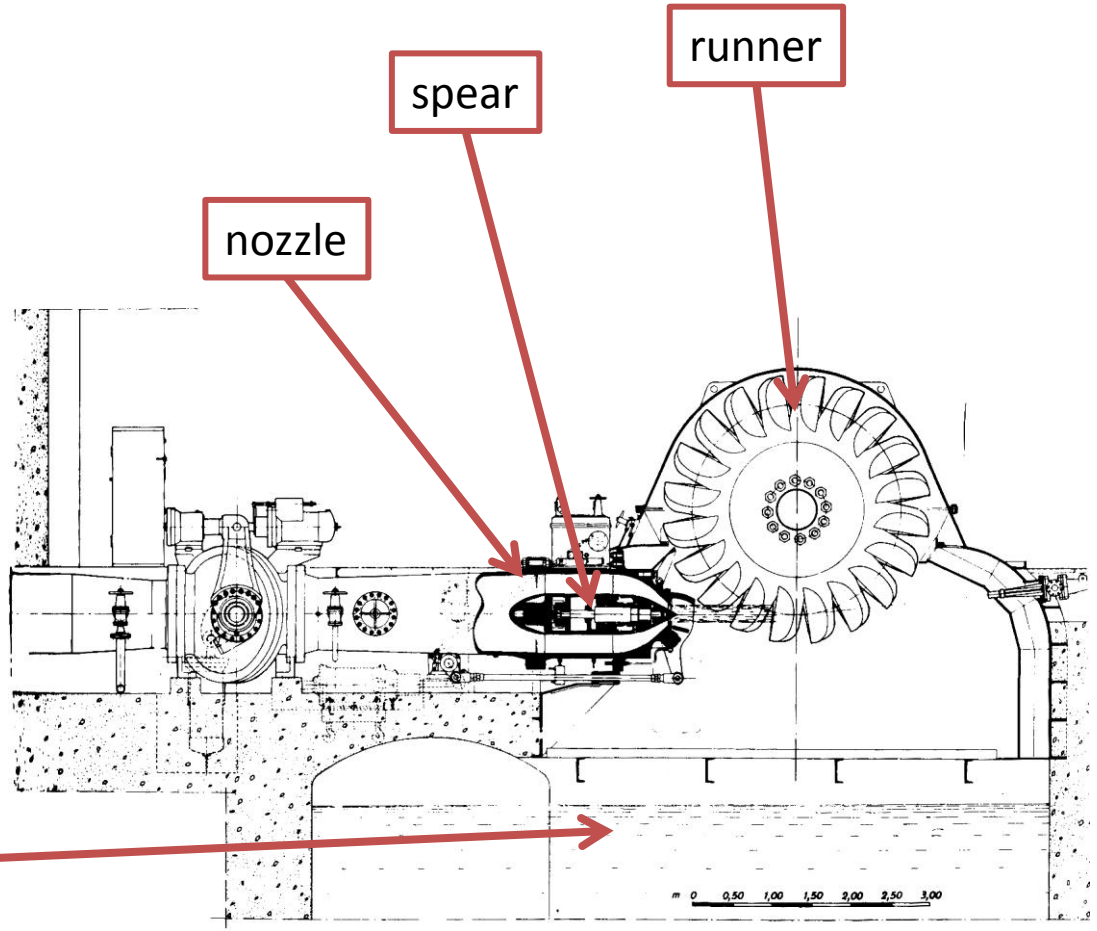


Source:
Pardeep Kumar, R.P. Saini, *Study of cavitation in hydro turbines—A review*, Renewable and Sustainable Energy Reviews, Volume 14, Issue 1, January 2010, Pages 374-383, ISSN 1364-0321, 10.1016/j.rser.2009.07.024.
(http://www.sciencedirect.com/science/article/pii/S1364032109001609)

Pelton Turbine

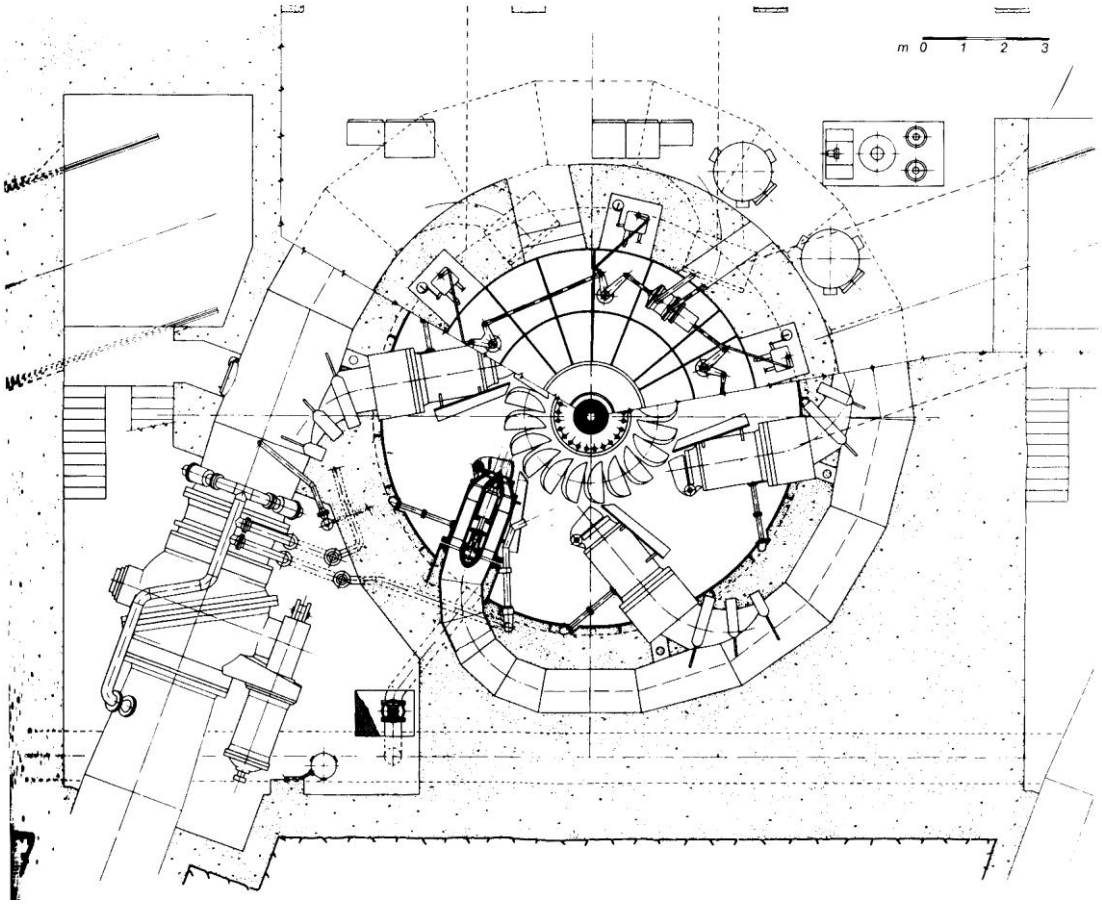
Horizontal axis 1-jet turbine





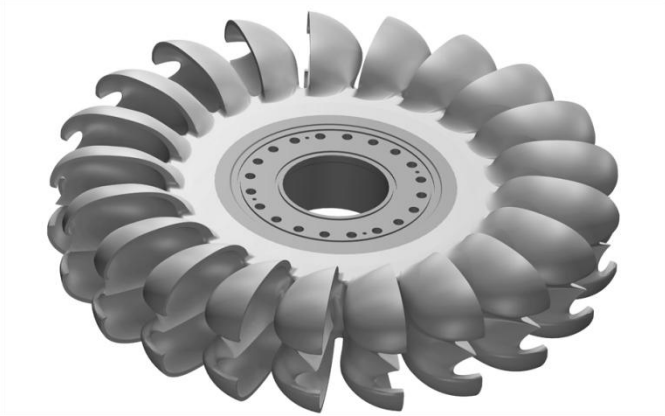
Pelton Turbine

Vertical axis, multiple jet turbine

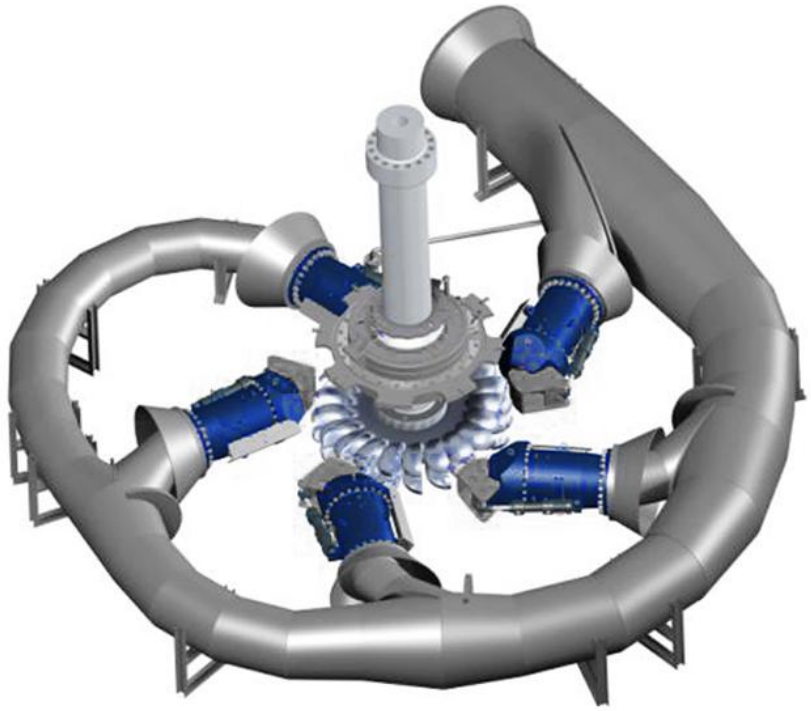


Pelton Turbine

Components



Runner



5-jet Pelton turbine

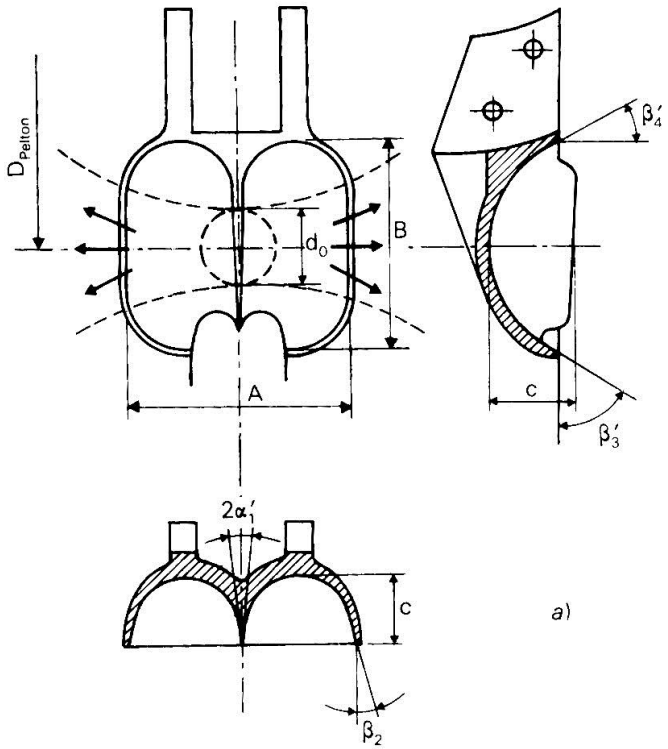
Source: Voith-Siemens

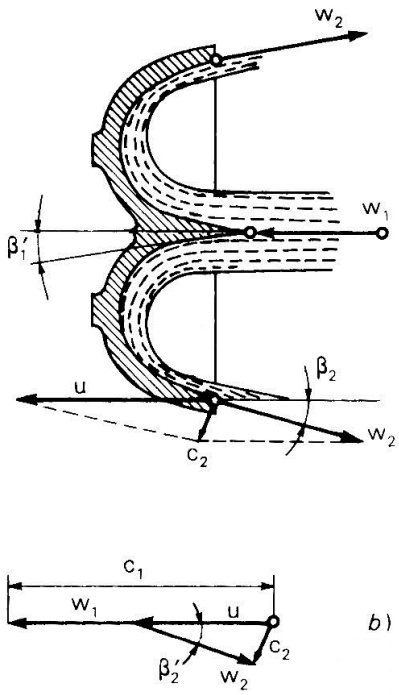
Pelton Turbine

5-jet turbine

Pelton Turbine

Bucket characteristics and velocity triangles





Pelton Turbine

Performance analysis

In case of ideal behavior inside the nozzle (no friction), water is discharged with a velocity given by:

$$c_{1,th} = \sqrt{2gH}$$

since kinetic energy in the upstream reservoir is negligible.

Friction inside the nozzle is taken into account by means of a **nozzle friction coefficient $\boldsymbol{\varphi}$**:

$$\boldsymbol{c_1 = \varphi\sqrt{2gH}}$$

Impulse turbine -> water does not accelerate in the runner -> relative velocity changes only because of friction, which is taken into account by means of a **runner friction coefficient $\boldsymbol{\psi}$**:

$$\boldsymbol{w_2 = \psi\, w_1}$$

Work per unit mass is given by Euler equation (u is the blade speed):

$$W = u\,(c_{1u} - c_{2u}) = u\,(c_1 - u - w_{2u}) = u\,(c_1 - u + \psi w_1 \cos\beta_2) = u\,[c_1 - u + \psi(c_1 - u)\cos\beta_2]$$

or

$$\boldsymbol{W = u(c_1 - u)(1 + \psi\, cos\, \beta_2)}$$

Pelton Turbine

Performance analysis

Hydraulic efficiency is given by:

$$\eta_y = \frac{W}{gH} = \frac{u(c_1 - u)(1 + \psi \cos\beta_2)}{\frac{c_1^2}{2\varphi^2}} = 2\varphi^2(1 + \psi \cos\beta_2)\frac{u}{c_1}\left(1 - \frac{u}{c_1}\right)$$

Maximum efficiency is obtained if $\boldsymbol{u/c_1 = 0,5}$:

$$\boldsymbol{\eta_{y,\max} = \frac{1}{2}\varphi^2(1 + \psi \cos\beta_2)}$$

Another way of maximizing efficiency would be given by setting $\beta_2 = 0$, but in order to avoid that water leaving the blade could strike the back of the following bucket it is necessary to have $\beta_2 > 0$. Usual values of blade angle at the exit are $\boldsymbol{\beta_2 = 10 \div 15°}$.

Power output is given by:

$$\boldsymbol{P = \eta_v \eta_y \rho Q g H = \eta_v\, \rho Q\, gH\, 2\varphi^2(1 + \psi \cos\beta_2)\frac{u}{c_1}\left(1 - \frac{u}{c_1}\right)}$$

while **torque** is:

$$\boldsymbol{C = \frac{P}{\omega} = \eta_v\, \rho Q \sqrt{gH}\, D \frac{\varphi}{\sqrt{2}}(1 + \psi \cos\beta_2)\left(1 - \frac{u}{c_1}\right)}$$

being $\omega = 2\pi n = 2u/D$.

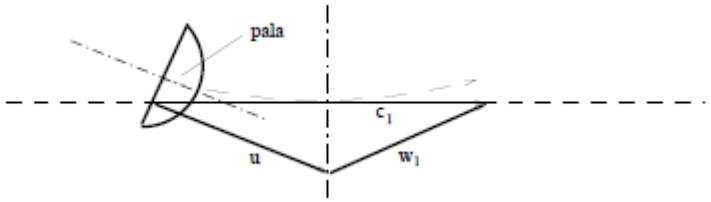
Pelton Turbine

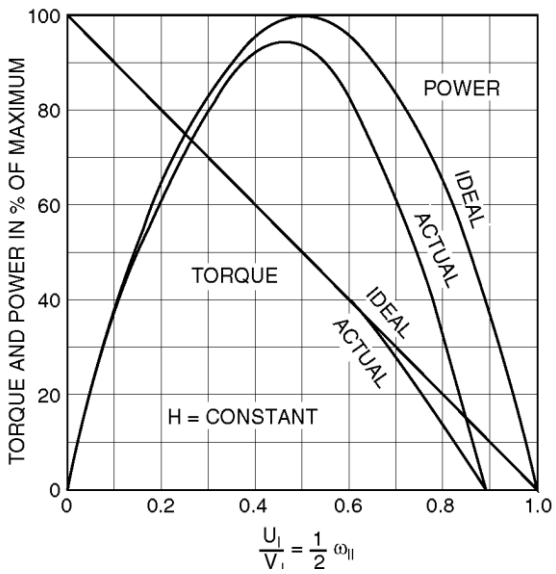
Performance analysis

The equations for efficiency, power and torque neglect two factors that decrease power and efficiency:

- “fan effect”: those buckets that are not struck by the jet actually behave like fan blades, “moving” the surrounding air, causing power losses proportional to u^3;
- water jet is not always perpendicular to the bucket, so relative velocity is higher and strikes the blade at a different angle than in the design configuration.

As a consequence, efficiency and torque differ (slightly) from their ideal behavior, as shown by this performance map.





Performance map taken from: R.E.A. Arndt, *Hydraulic turbines*, in *The Engineering Handbook – Second Edition*, chapter 73, CRC Press LLC, 2005.

Pelton Turbine

Performance analysis

Volumetric flow rate depends on **jet diameter d_j**, **nozzle exit velocity c_1** and **number of jets N_j**:

$$\boldsymbol{Q = N_j \frac{\pi}{4} d_j^2 c_1}$$

Thus, flow rate depends on head (through c_1), while it is not affected by rotational speed:

$$Q = N_j \frac{\pi}{4} d_j^2 \varphi \sqrt{2gH} \Rightarrow \boldsymbol{Q \propto N_j d_j^2 \sqrt{H}}$$

For a Pelton turbine, specific speed k is thus given by:

$$k \propto \omega \, d_j \sqrt{\frac{N_j}{H}}$$

$$\omega \propto \frac{u}{D} \propto \frac{1}{D}\left(\frac{u}{c_1}\right) c_1 \propto \frac{1}{D}\left(\frac{u}{c_1}\right)\sqrt{H}$$

$$\boldsymbol{k \propto \frac{d_j}{D} \sqrt{N_j}}$$

The proportionality constant can be taken as approximately 1.3:

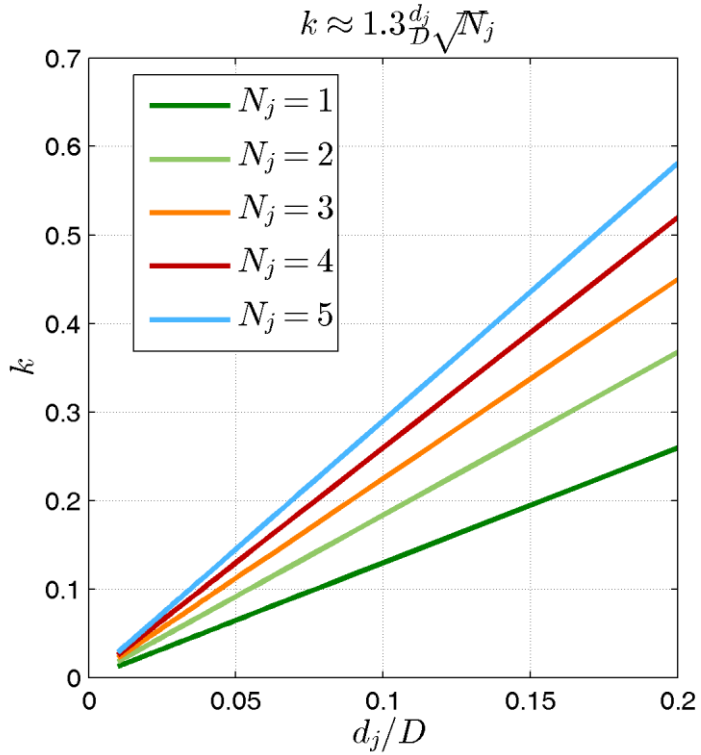
$$\boldsymbol{k \approx 1.3 \frac{d_j}{D} \sqrt{N_j}}$$

Pelton Turbine

Size of the turbine

The relationship among k, N_j and the ratio d_j/D gives an indication on the required machine size, taking into account design parameters such as:

- gross head -> net head
- flow rate
- rotational speed -> wheel diameter



Example:		
	rotational speed	50 s^{-1}
	volumetric flow rate	2 m^3/s
	net head	1500 m
	ratio u/c_1	0.48
	nozzle friction coeff. φ	0.98

results:

$$k = 0.33 \Rightarrow N_j = 5$$

$$c_1 \cong 168 \text{ m/s}$$

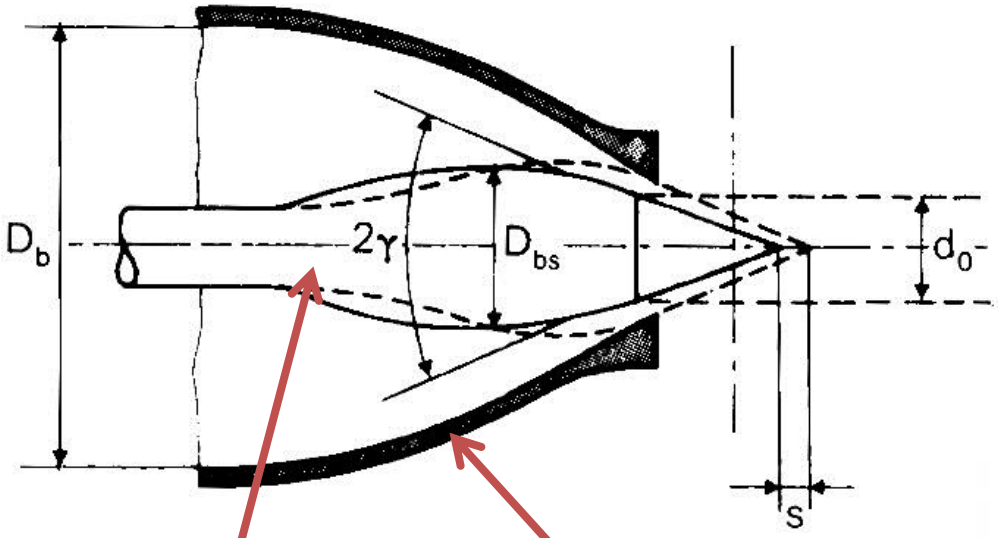
$$u \cong 80.7 \text{ m/s}$$

$$D \cong 0.514 \text{ m}$$

$$d_j \cong 0.055 \text{ m} \Rightarrow \frac{d_j}{D} \cong 0.107 \qquad \text{OK}$$

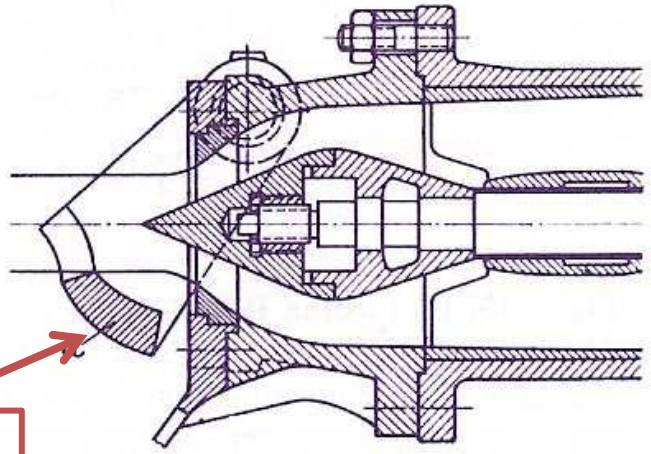
Pelton Turbine

Flow rate and power control



spear (needle)

nozzle



deflector

Pelton Turbine

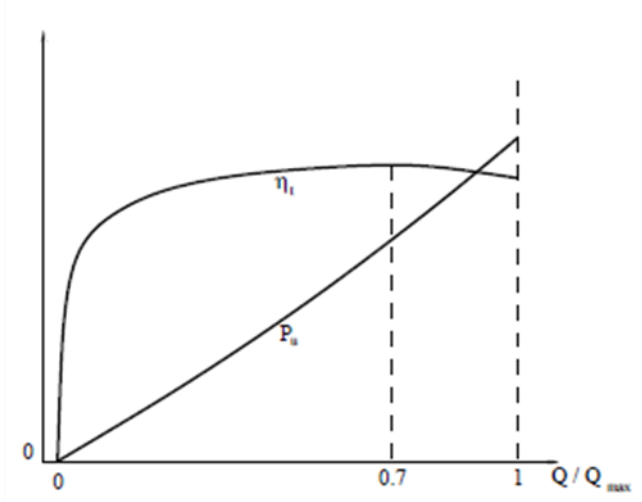
Power control

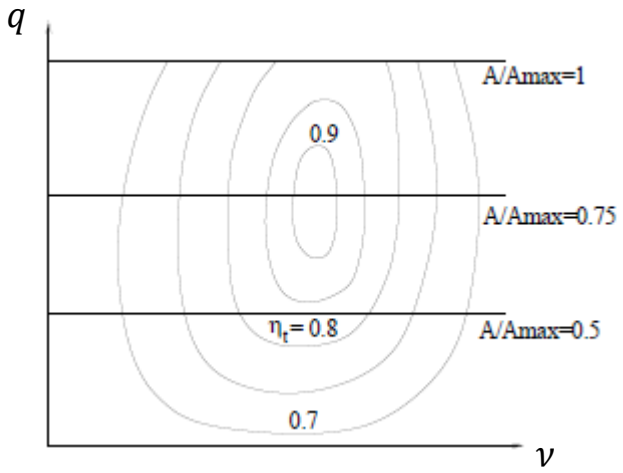
Jet velocity (and thus work output) is only marginally affected by flow rate, through the nozzle friction coefficient φ.

Main factors that influence performance at part load:

1. power loss due to friction in the nozzle is almost constant -> reduction of coefficient φ
2. power loss due to friction in the runner is almost constant too -> reduction of coefficient ψ
3. as jet diameter decreases, it no longer perfectly matches blade profiles -> kinetic energy losses at runner exit increase
4. power losses due to “fan effect” do not depend on flow rate -> its relative importance thus grows as power output decreases

Anyway, **Pelton turbines behave very well under part load operating conditions**.

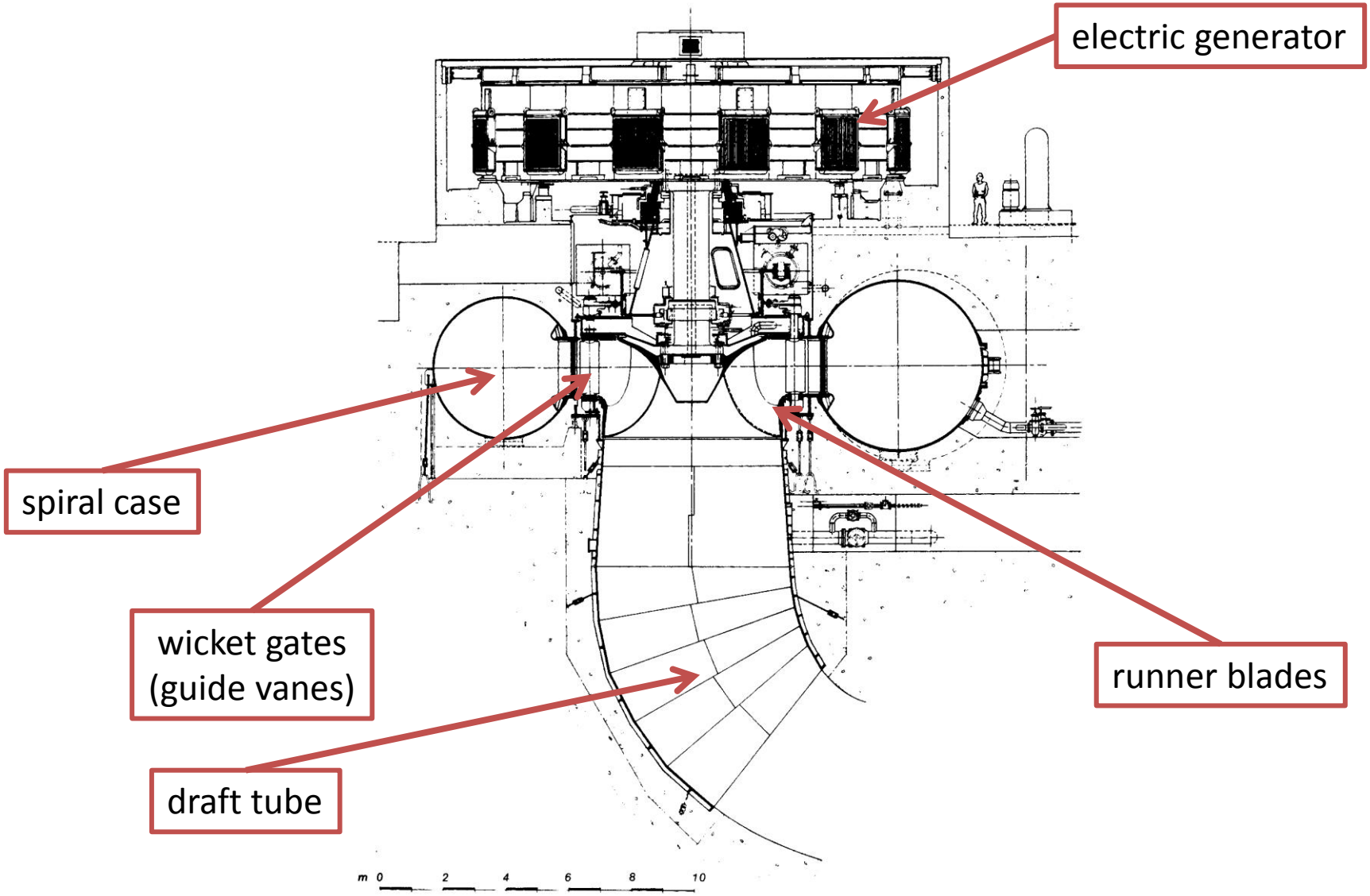




Performance curves taken from: M. Napolitano, P. De Palma, G. Pascazio, *Turbine idrauliche* , dispense per il corso di Macchine, Politecnico di Bari

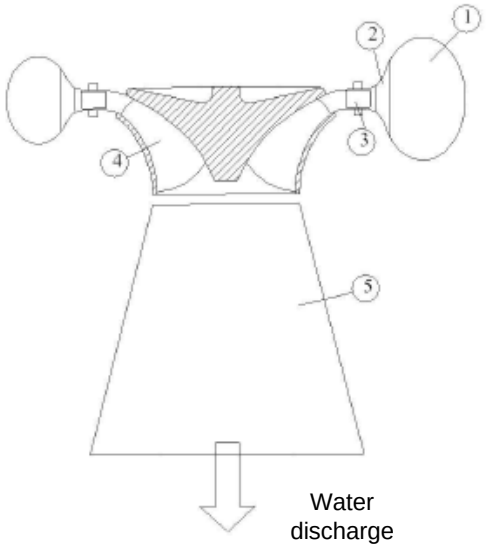
Francis Turbine

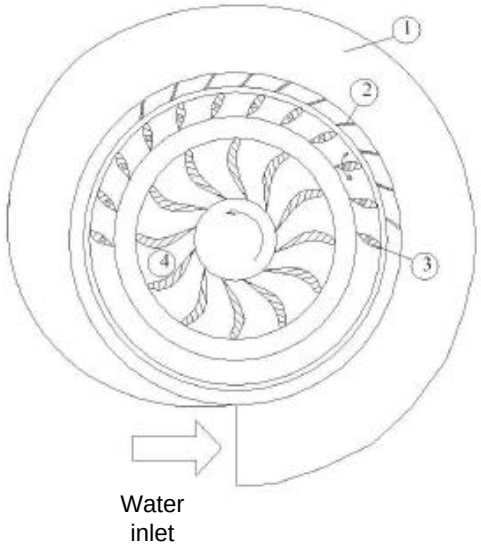
Main components



Francis Turbine

Main components





1. spiral case
2. stay vanes
3. wicket gates (guide vanes)
4. runner
5. draft tube

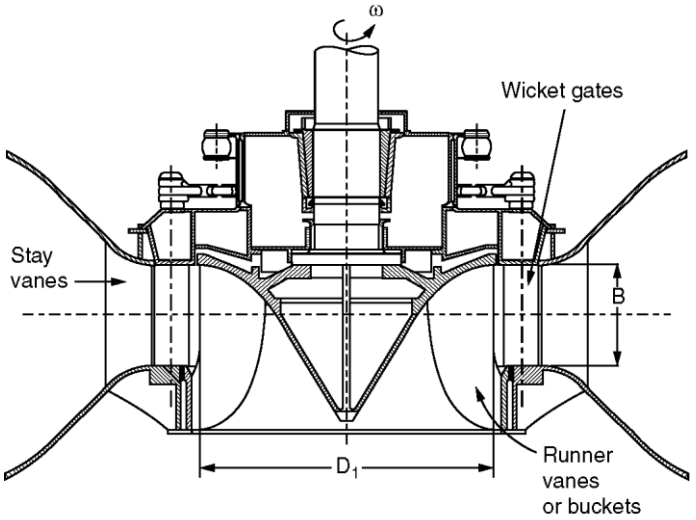
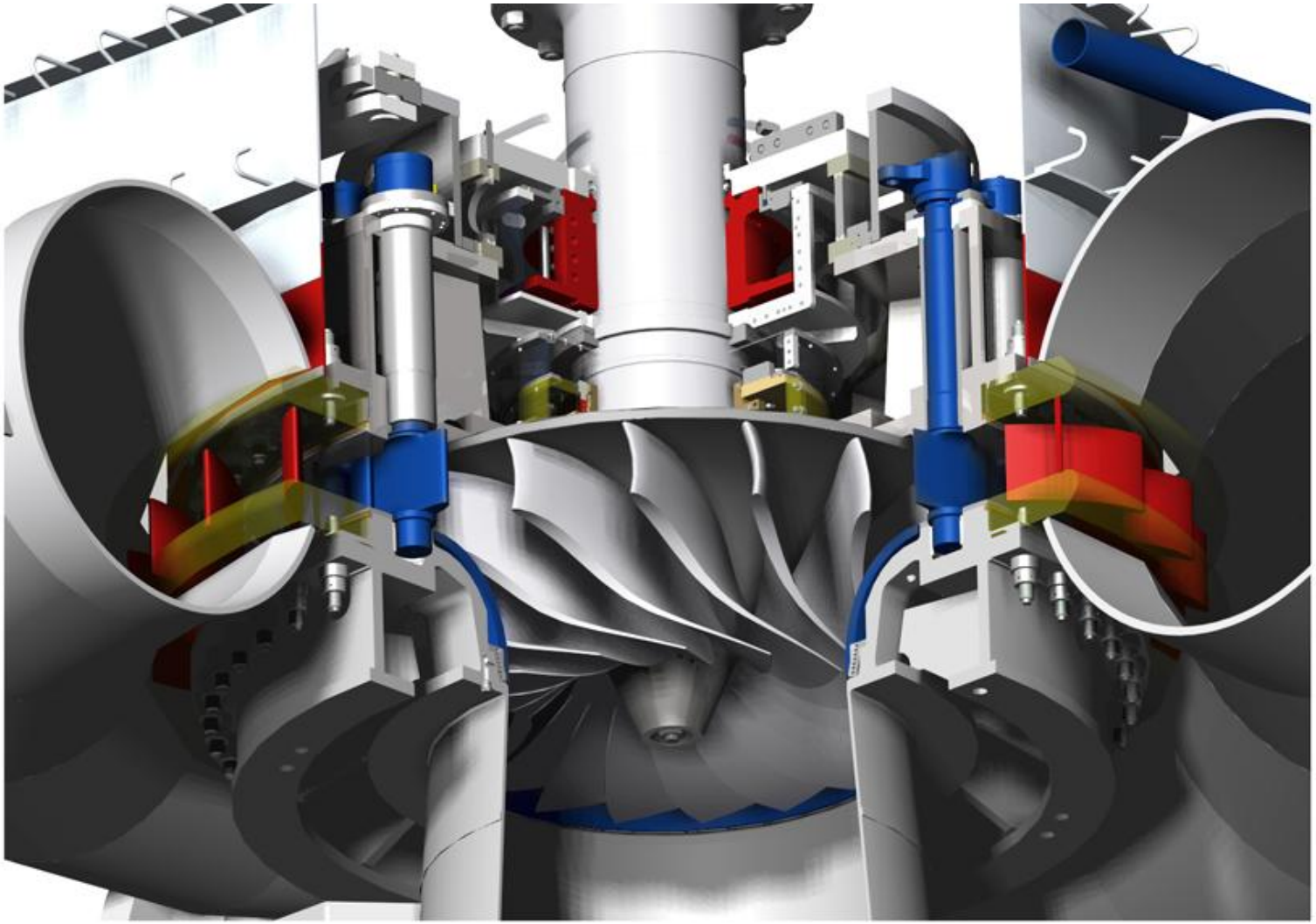


Figure (lower right) taken from: R.E.A. Arndt, *Hydraulic turbines*, in *The Engineering Handbook – Second Edition*, chapter 73, CRC Press LLC, 2005.

Francis Turbine

Main components



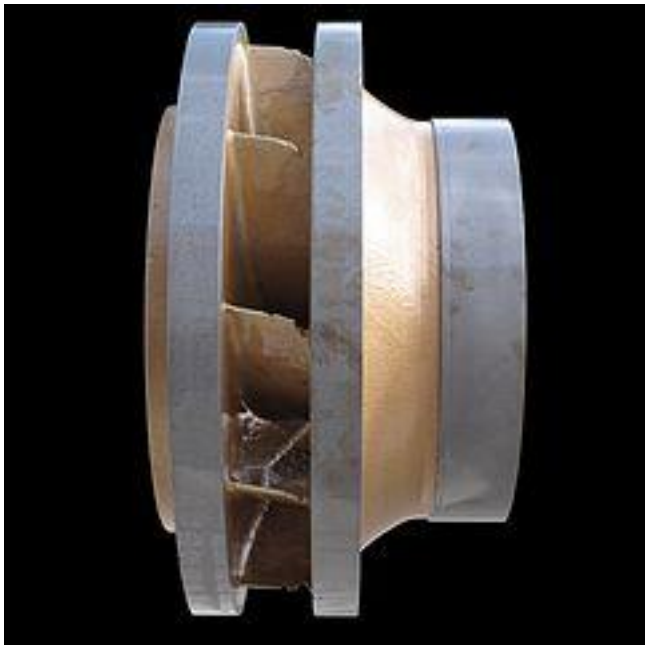
Source: Voith-Siemens

Francis Turbine

Main components

Francis Turbine

Runner



1956 Francis turbine runner
power output 410 kW
rotational speed 1000 rpm

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Francis Turbine

Runner

Grand Coulee Dam (USA)
Nominal net head 116 m

Three Gorges Dam, People's Republic of China
Nominal power output 700 MW
Nominal net head 80.6 m

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Francis Turbine

Influence of specific speed on runner blade configuration

Since

$$n_s \propto \sqrt{\frac{l_1}{D_1}\left(\frac{1}{1-R}\right)^{1/2} \tan\alpha_1}$$

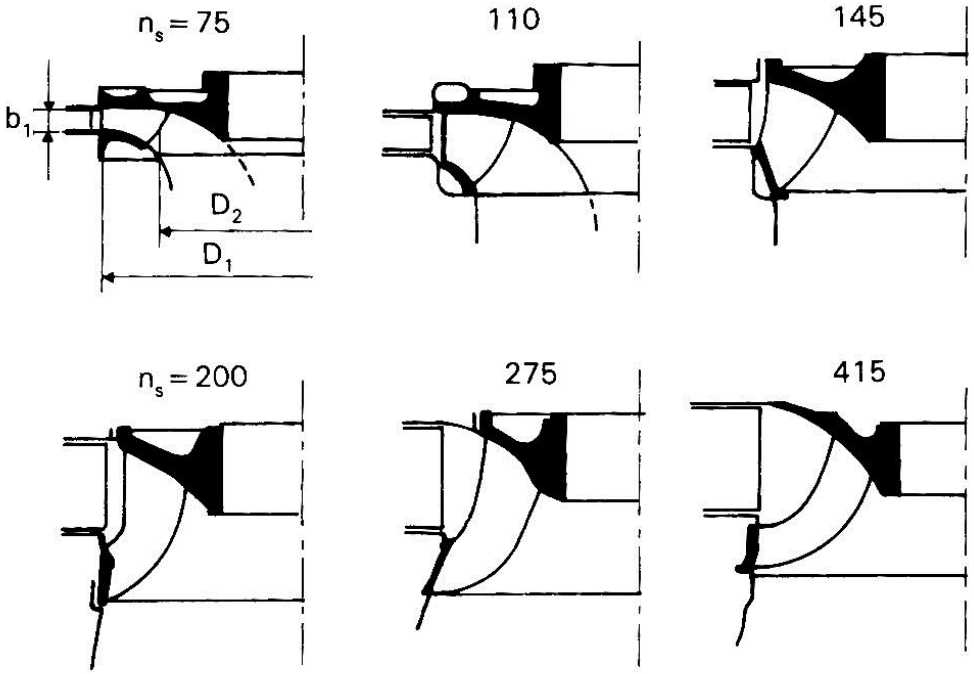
the ratio l_1/D_1 increases as specific speed increases. The same holds true for stage reaction and inlet direction α_1.

Furthermore, if $c_{1m} \cong c_{2m}$:

$$l_1 D_1 \cong \frac{D_2^2}{4} \Rightarrow \frac{D_2}{D_1} \cong 2\sqrt{\frac{l_1}{D_1}}$$

Turbines with a small gap between wicket gates and runner $\rightarrow D_2 < D_1 \rightarrow$ low specific speed n_s ("slow" turbine).

In order for the specific speed to increase, exit diameter must become larger than at the inlet ($D_2 > D_1$), which can be done if the configuration goes toward axial flow in the runner ("fast" turbine).

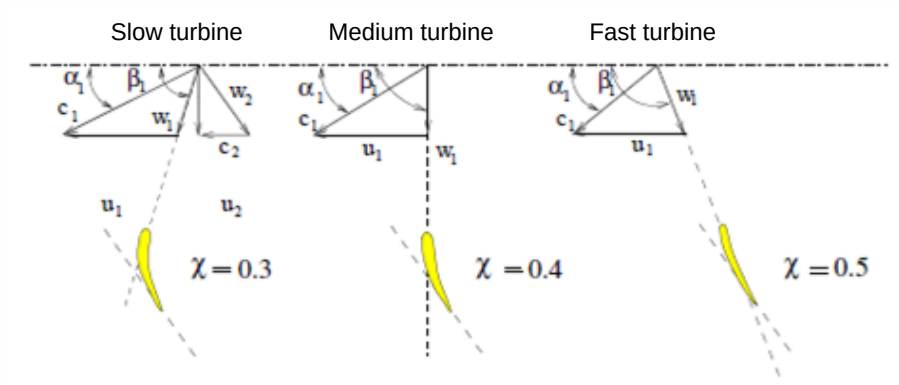


Francis Turbine

Influence of specific speed on runner blade configuration

$$n_s \propto \sqrt{\frac{l_1}{D_1}\left(\frac{1}{1-R}\right)^{1/2} \tan\alpha_1}$$

In order to achieve high values of specific speed, both stage reaction and inlet directions α_1 and β_1 must increase.



	n_s	R	α_1	β_1
Slow turbines	15 ÷ 33	0,30	15 ÷ 20°	60 ÷ 70°
Medium turbines	33 ÷ 55	0,40	25 ÷ 30°	~ 90°
Fast turbines	55 ÷ 80	0,50	35 ÷ 40°	120 ÷ 130°

Image source: M. Napolitano, P. De Palma, G. Pascazio, *Turbine idrauliche* , dispense per il corso di Macchine, Politecnico di Bari

Francis Turbine

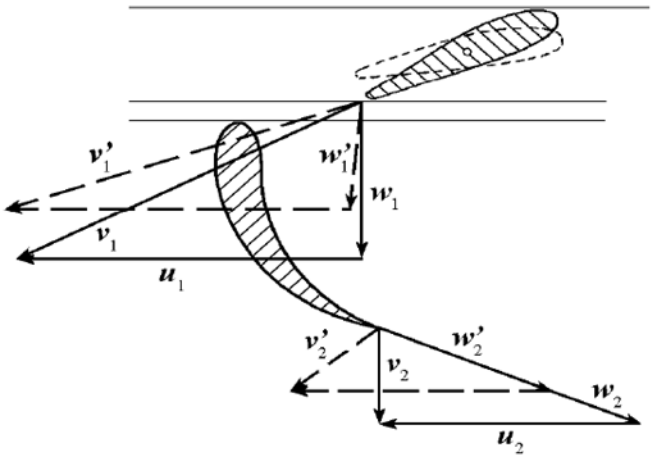
Flow rate control: adjustable wicket gate blades

For Francis turbines, **flow rate** (and thus power output) **is controlled by changing the inclination of wicket gate blades**.

This allows to reduce the radial component of water velocity.

A distinctive disadvantage of this kind of power control is that, under part load operating conditions, **water approaches the runner with a different direction with respect to the design direction β_1**, with a corresponding **impact loss** due to the mismatch between water direction and blade profile.

Furthermore, water velocity at runner exit gains a tangential component, therefore relatively increasing kinetic energy losses.



Francis Turbine

Flow rate control: adjustable wicket gate blades

Full opening

Minimum opening

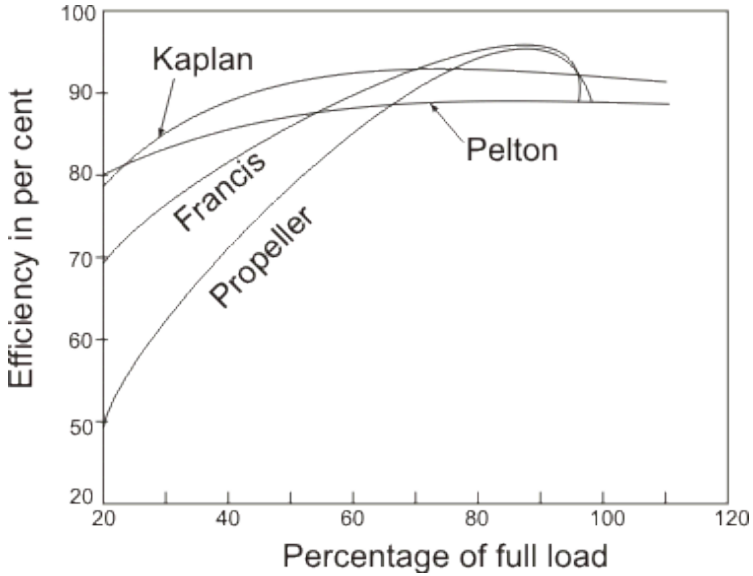
Francis Turbine

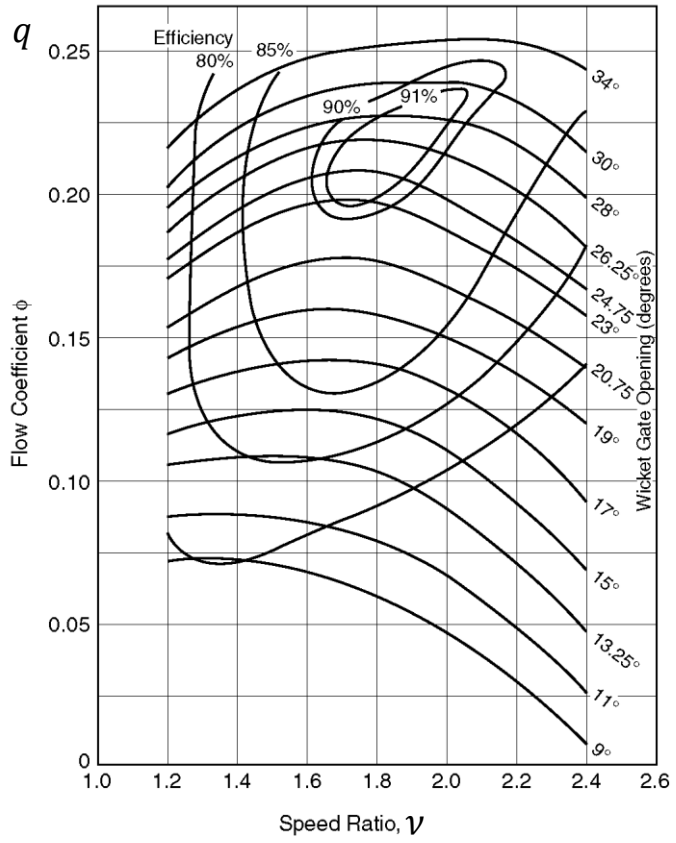
Efficiency

Part load operation is affected by **significant impact losses** so efficiency rapidly decreases as the operating point gets farther from design conditions.

Since “fast” turbines operate with higher flow rates and thus higher velocities, impact losses affect more substantially these turbines rather than “slow” ones.

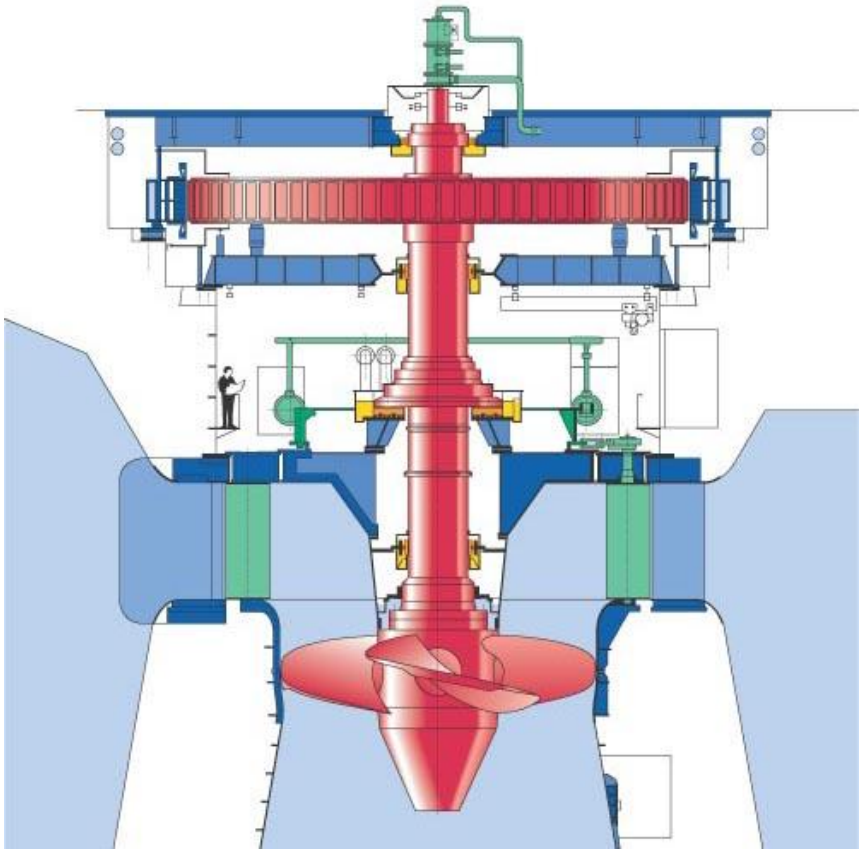
Anyway, **Francis turbines are much less suited to operate under variable operating conditions than Pelton turbines.**

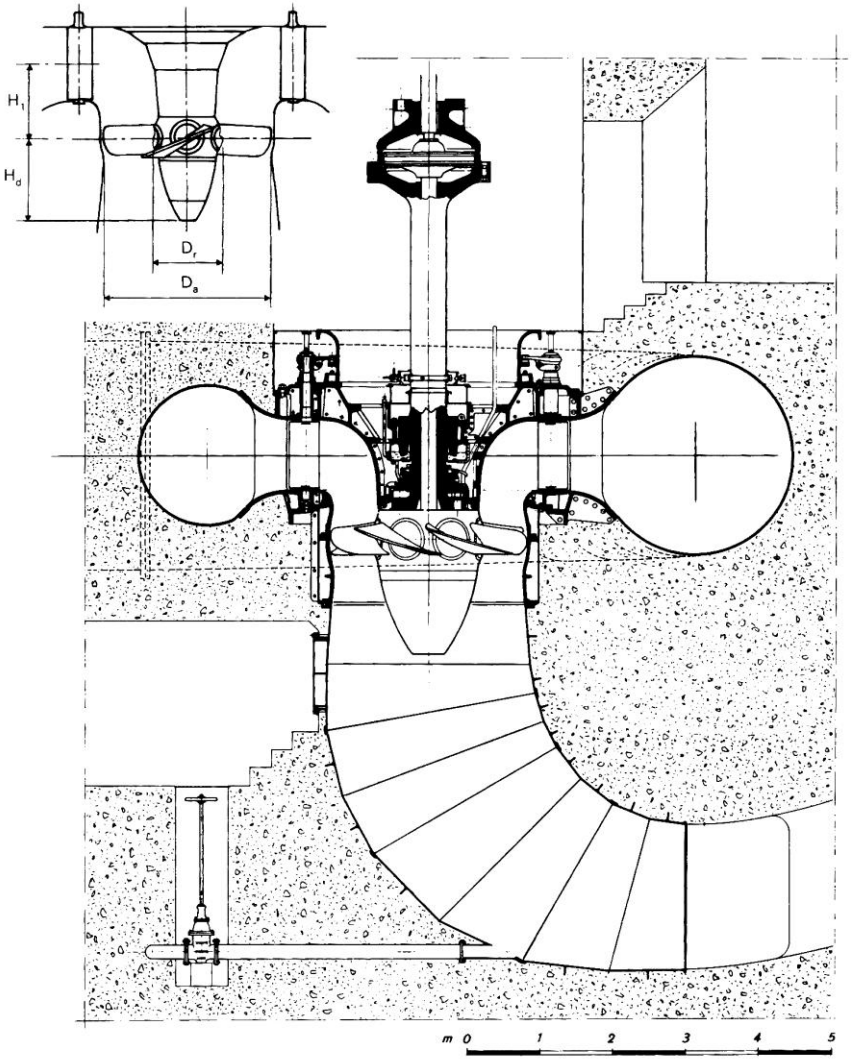




Kaplan Turbine

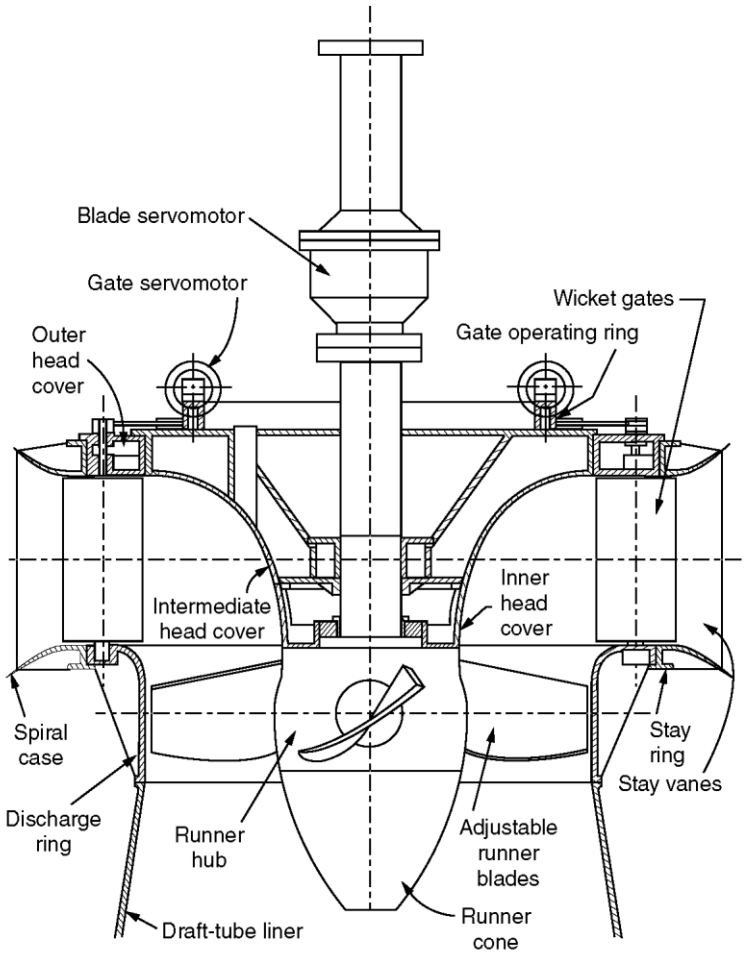
Turbine configuration

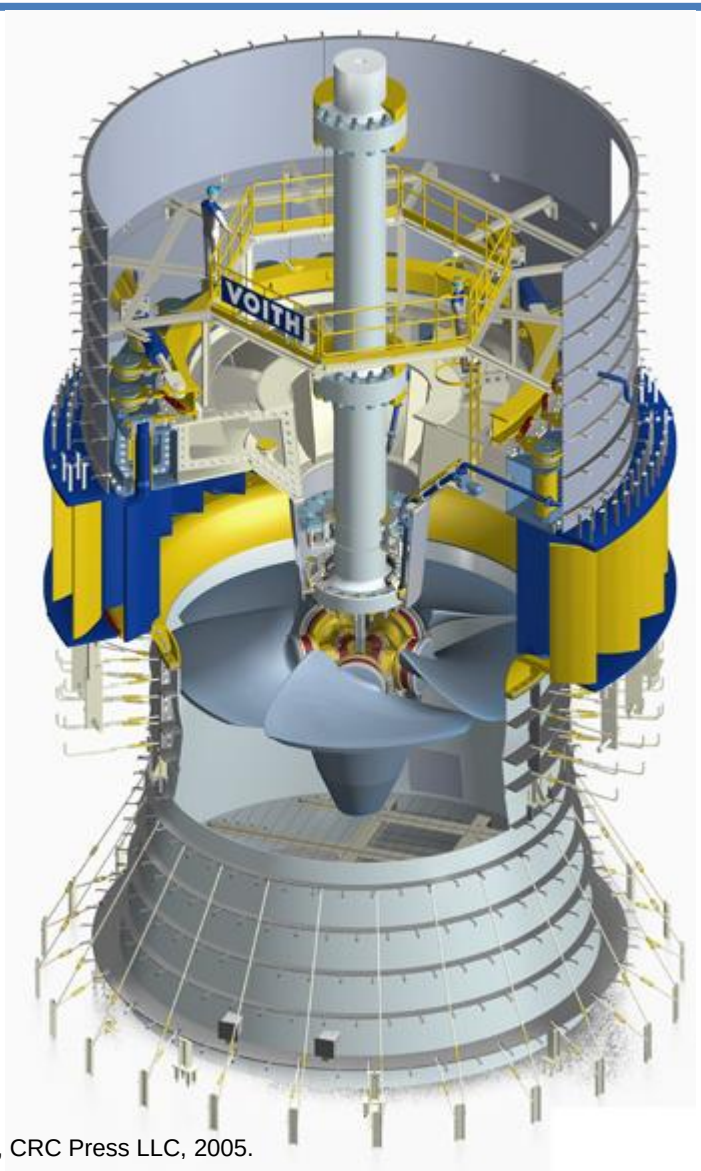




Kaplan Turbine

Turbine configuration





Source:
right: Voith-Siemens
left: R.E.A. Arndt, *Hydraulic turbines*, in *The Engineering Handbook – Second Edition*, chapter 73, CRC Press LLC, 2005.

Kaplan Turbine

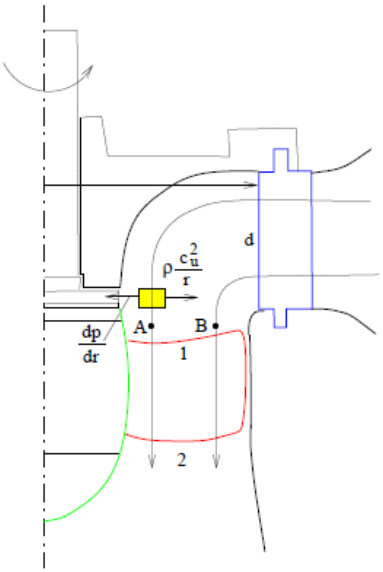
Velocity triangles

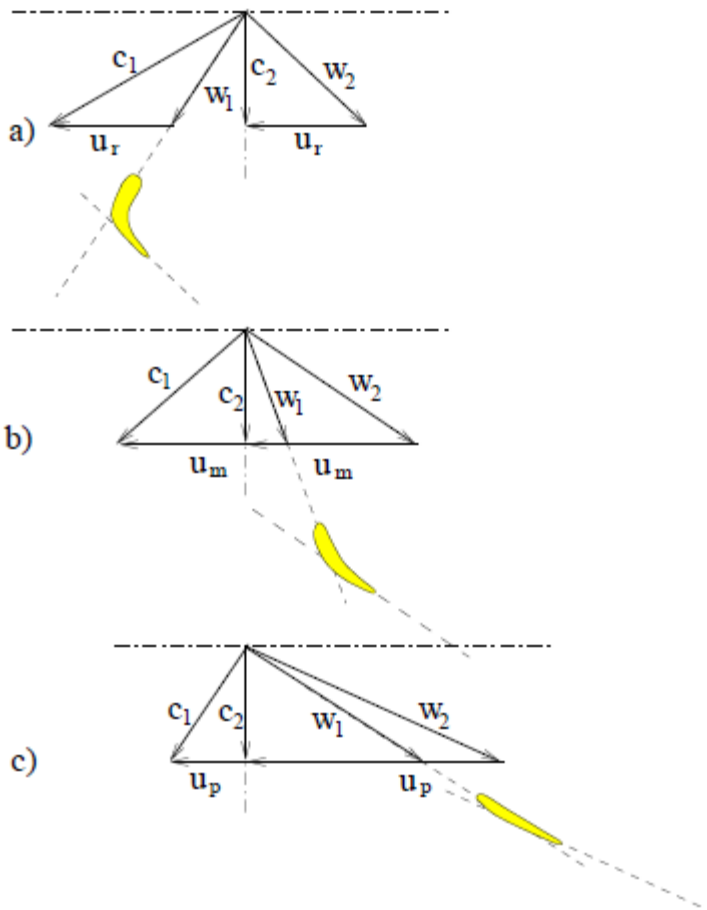
Inside the channel linking wicket gate (d) exit to runner inlet (1) there is nothing to guide the water, which thus flows according to a free-vortex motion:

$$c_u r = \text{cost.}$$
$$c_a = \text{cost.}$$

Furthermore, a pressure gradient in the radial direction arises.

If runner blades are twisted according to a free-vortex design, these flow characteristics persist while water flows through the runner.





Source: M. Napolitano, P. De Palma, G. Pascazio, *Turbine idrauliche* , dispense per il corso di Macchine, Politecnico di Bari

Kaplan Turbine

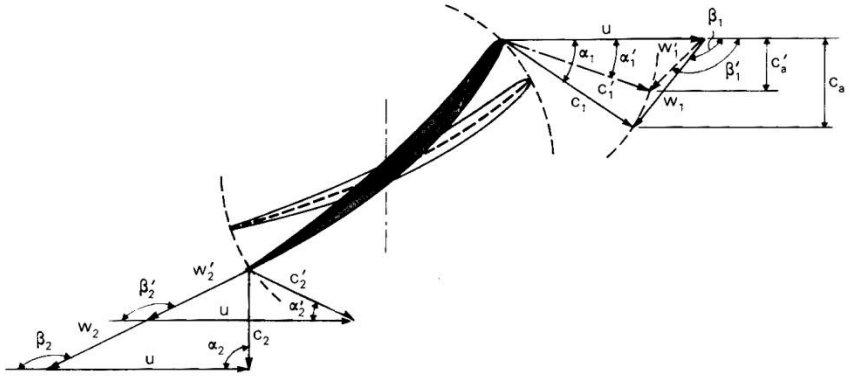
Flow rate control: adjustable wicket gate and runner blades

Mass and angular momentum conservation equations:

$$l_d D_d c_d \sin \alpha_d = l_1 D_1 c_1 \sin \alpha_1$$
$$r_d c_d \cos \alpha_d = r_1 c_1 \cos \alpha_1$$

In order to have runner blades correctly aligned with incoming water, the blade must be rotated in such a way that the following equation is satisfied:

$$\tan \alpha_1 = \frac{l_d}{l_1} \tan \alpha_d$$



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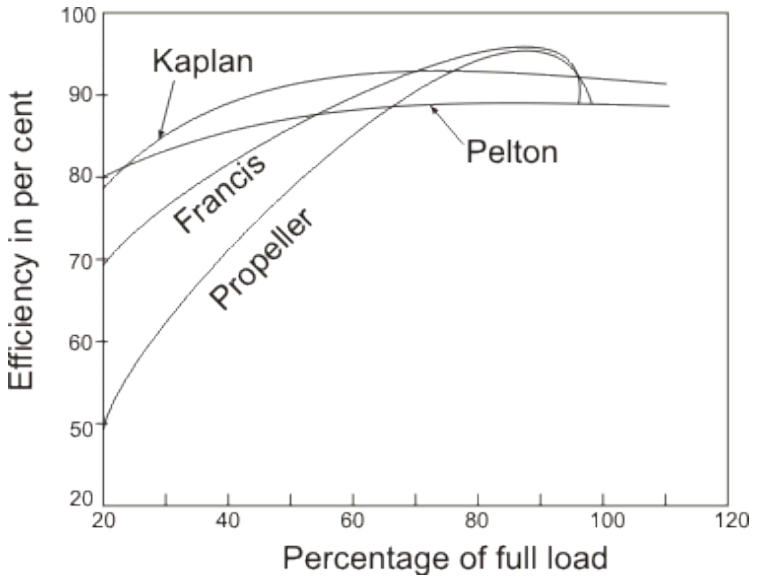
Kaplan Turbine

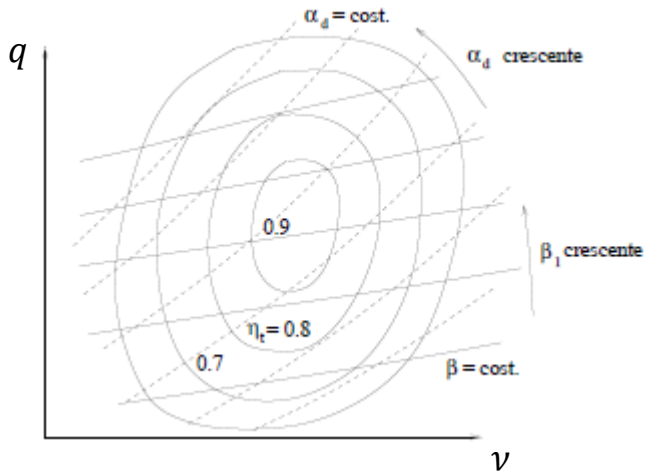
Efficiency

The variable-pitch runner blades allow **Kaplan turbines** to **achieve very high efficiencies even at part load operation** and for a wide range of power output, because impact losses are avoided.

In the case of **simple propeller turbines** (fixed pitch runner blades), heavy losses occur, as in the case of Francis turbines, and **the efficiency penalty is particularly pronounced due to relatively high water velocity**.

The diagram at the bottom illustrates a typical hill diagram for a Kaplan turbine.





Bulb Turbine

Turbine configuration

Bulb turbines take full advantage of the axial flow configuration: immersed in the water channel, the flow enters and exits the turbine with minor changes in direction.

Bulb turbines may have fixed pitch or variable pitch (Kaplan) blades, and different configurations are possible:

- **Bulb (tubular) turbine:** the bulb holds electric generator, wicket gates and runner.
- **Pit turbine:** a gear box is used in order to reduce generator and bulb size; the generator is not enclosed in the bulb.
- ***Straflo* (straight flow) turbine:** the rotor of the electric generator is directly connected to the runner, thus avoiding the need of a drive shaft, reducing the bulb size and increasing the flow area.
- **S-turbine:** the generator is placed outside the water channel by means of an S-shaped channel and a drive shaft connecting runner and generator .

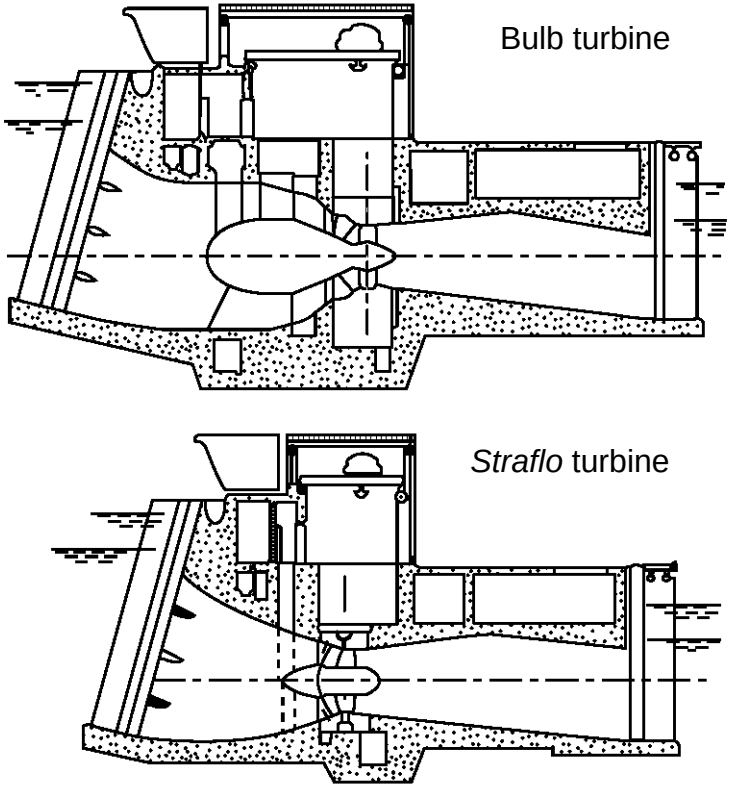
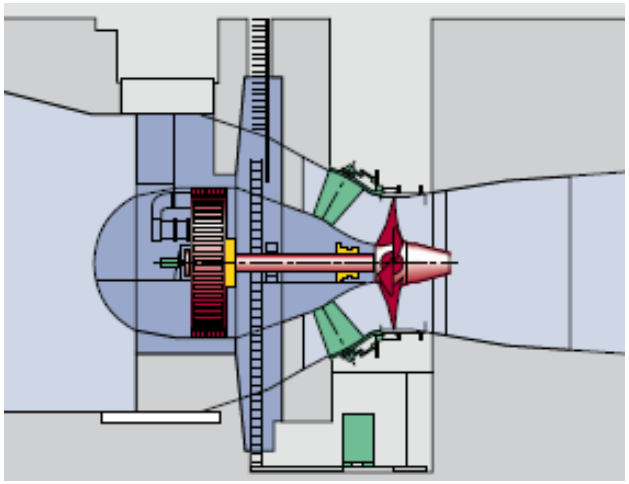


Image source: R.E.A. Arndt, *Hydraulic turbines*, in *The Engineering Handbook – Second Edition*, chapter 73, CRC Press LLC, 2005.

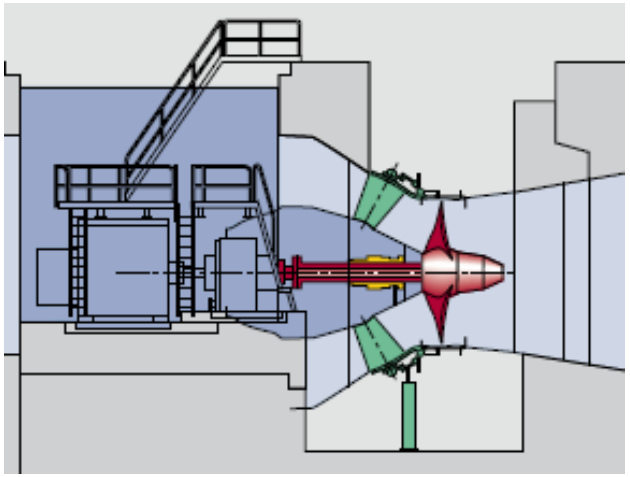
Bulb Turbine

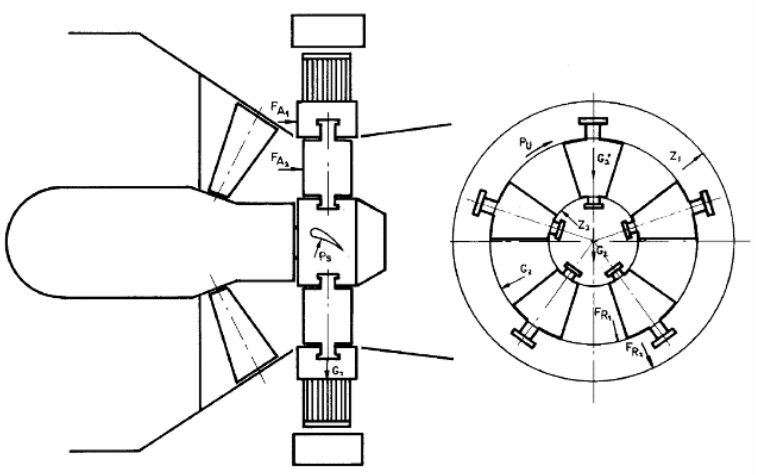
Turbine configuration

Bulb turbine

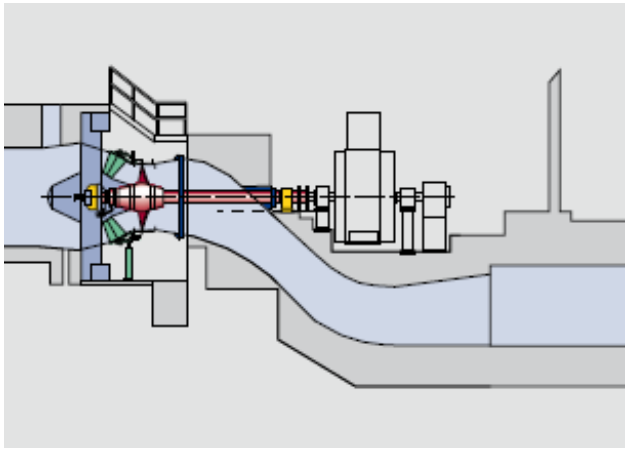


Pit turbine





Straflo turbine

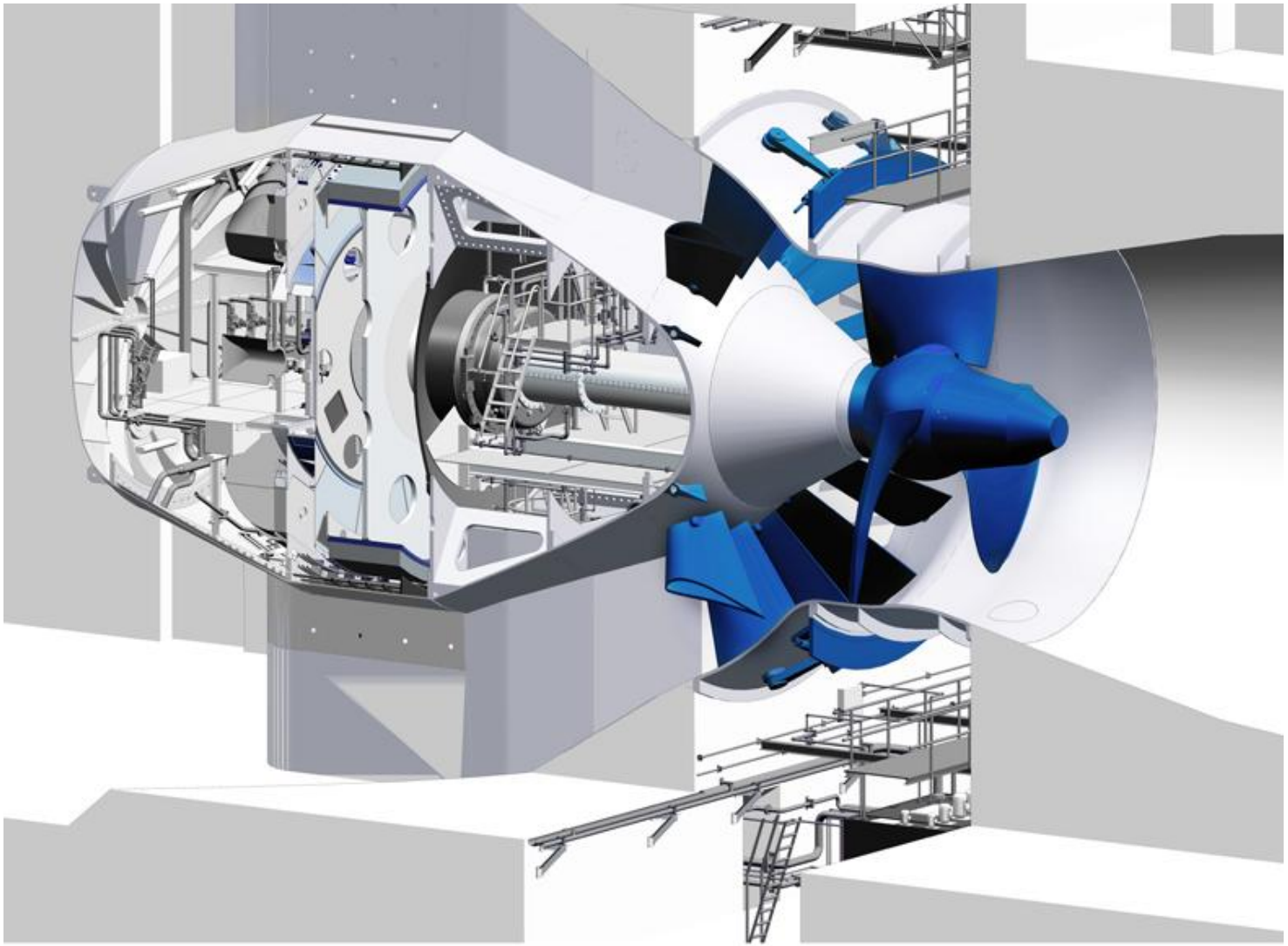


S-turbine

Source: Voith-Siemens

Bulb Turbine

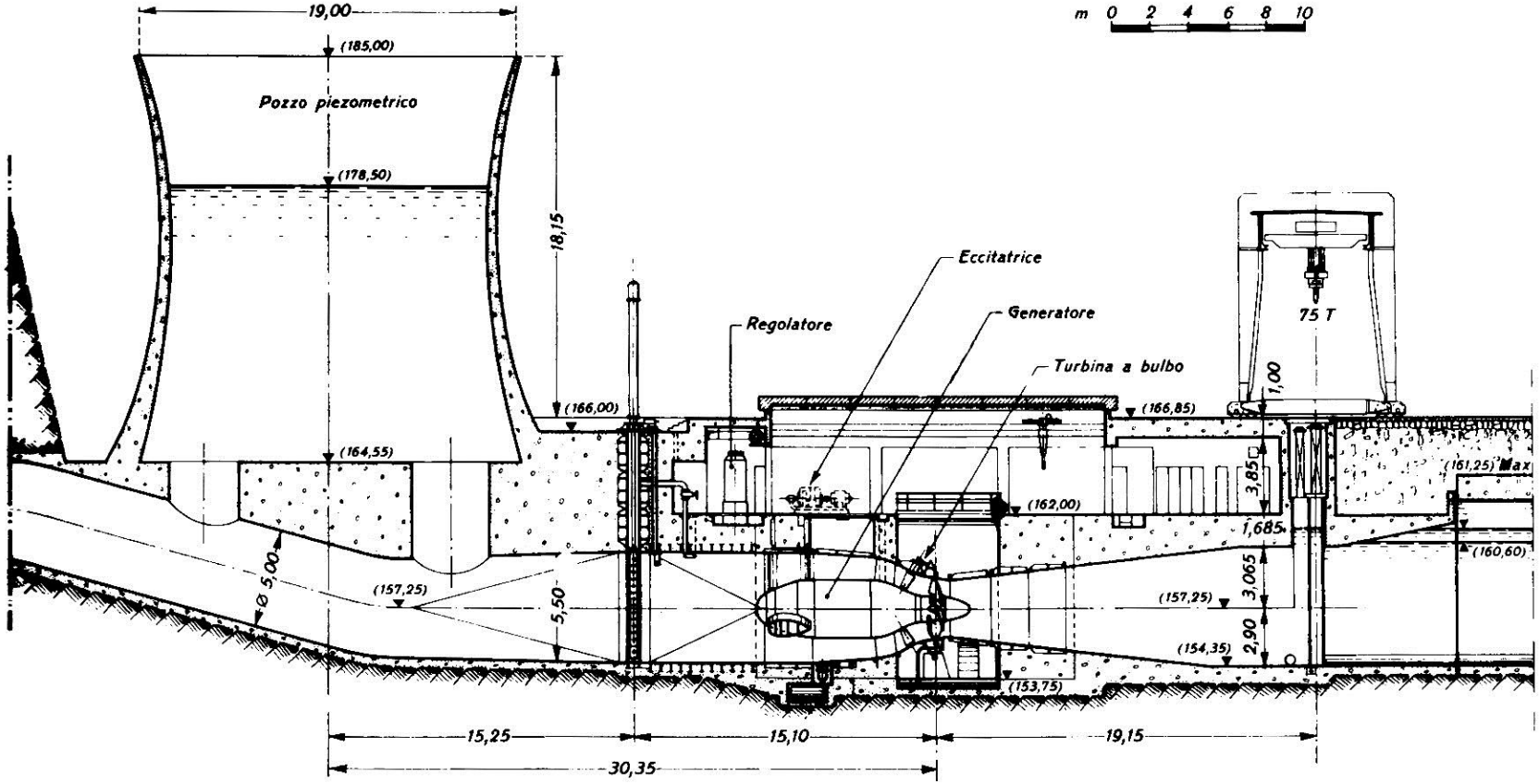
Turbine layout



Source: Voith-Siemens

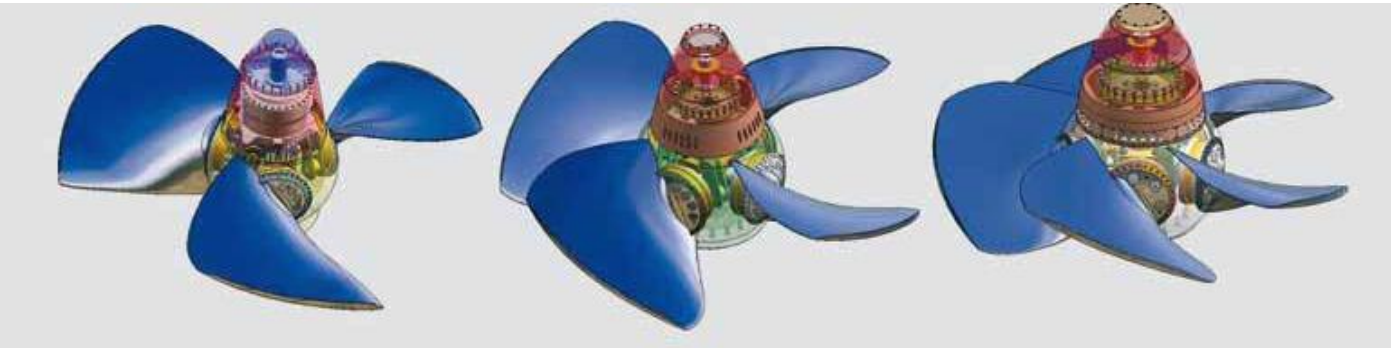
Bulb Turbine

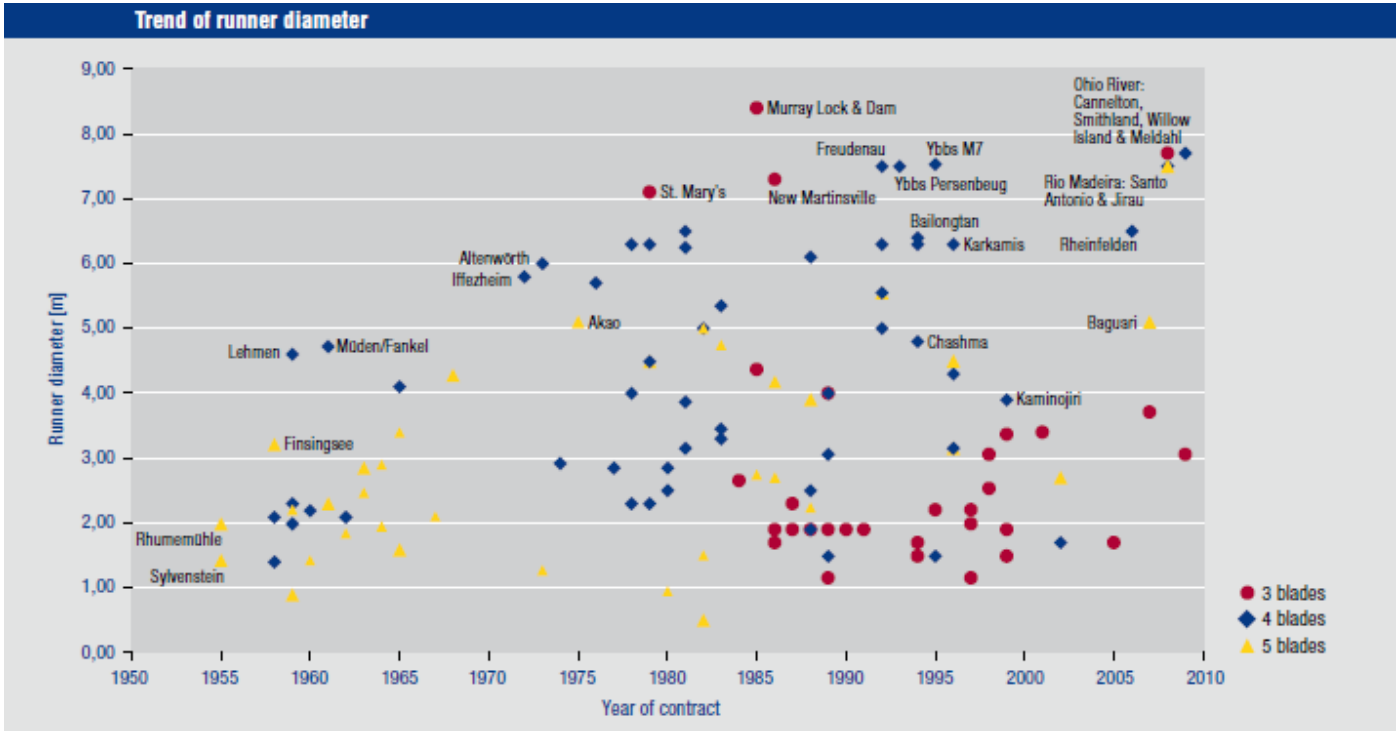
Plant layout



Bulb Turbine

Runner





Source: Voith-Siemens

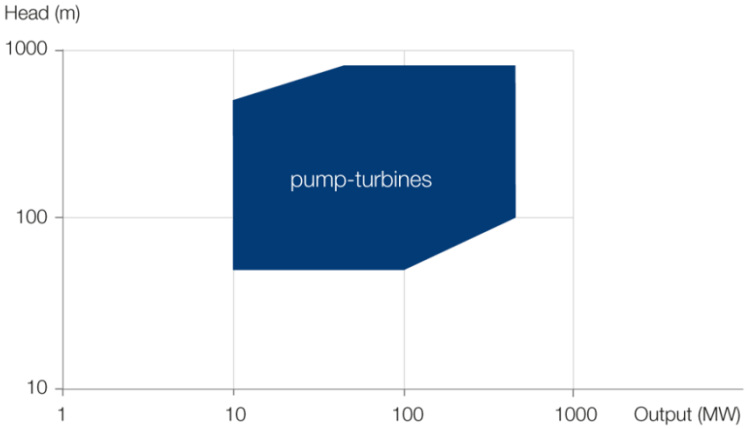
Pump Turbine (Reversible Turbine)

Main characteristics

Pump turbines are used in so-called ***pumped storage plants*** to transfer water to a high storage reservoir during off-peak hours.

These plants, therefore, are useful for smoothing out the difference between energy demand and supply: they can favorably **store energy produced by base-load plants during off-peak hours while making this energy available to the grid for peaking supply needs and system regulation**.

Pump turbines are used in a wide range of situations, with heads from less than 50 m to over 800 m, and unit power from 10 to over 500 MW.



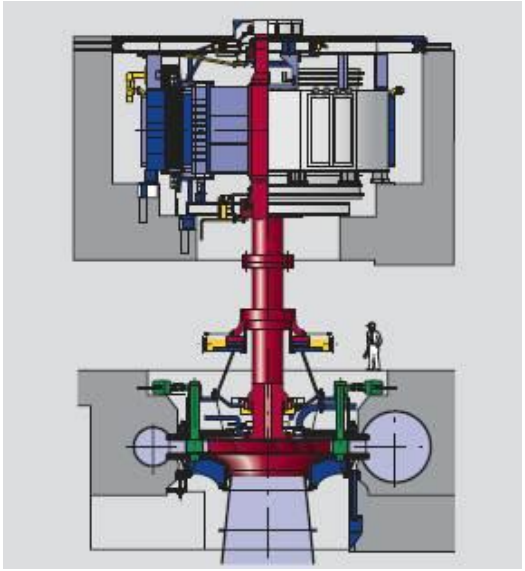
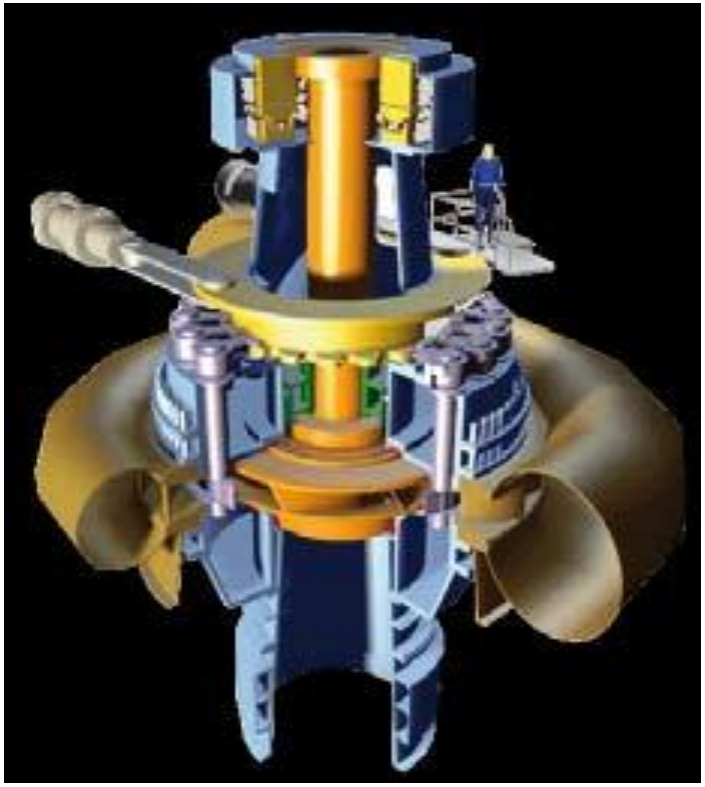


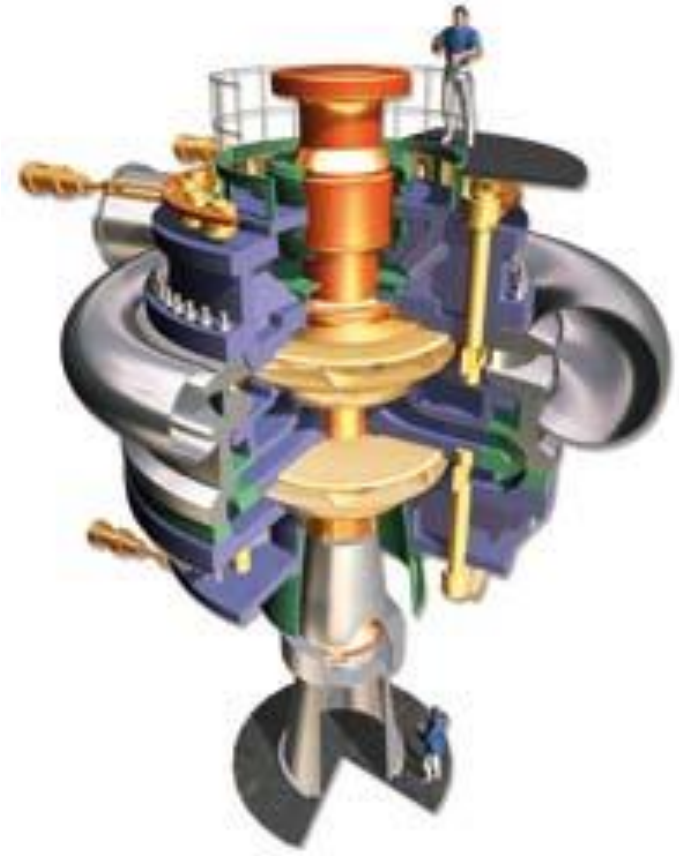
Image source: Voith-Siemens

Pump Turbine

Single-stage vs. double- or multi-stage centrifugal units



Single-stage pump turbine (H < 700 m)

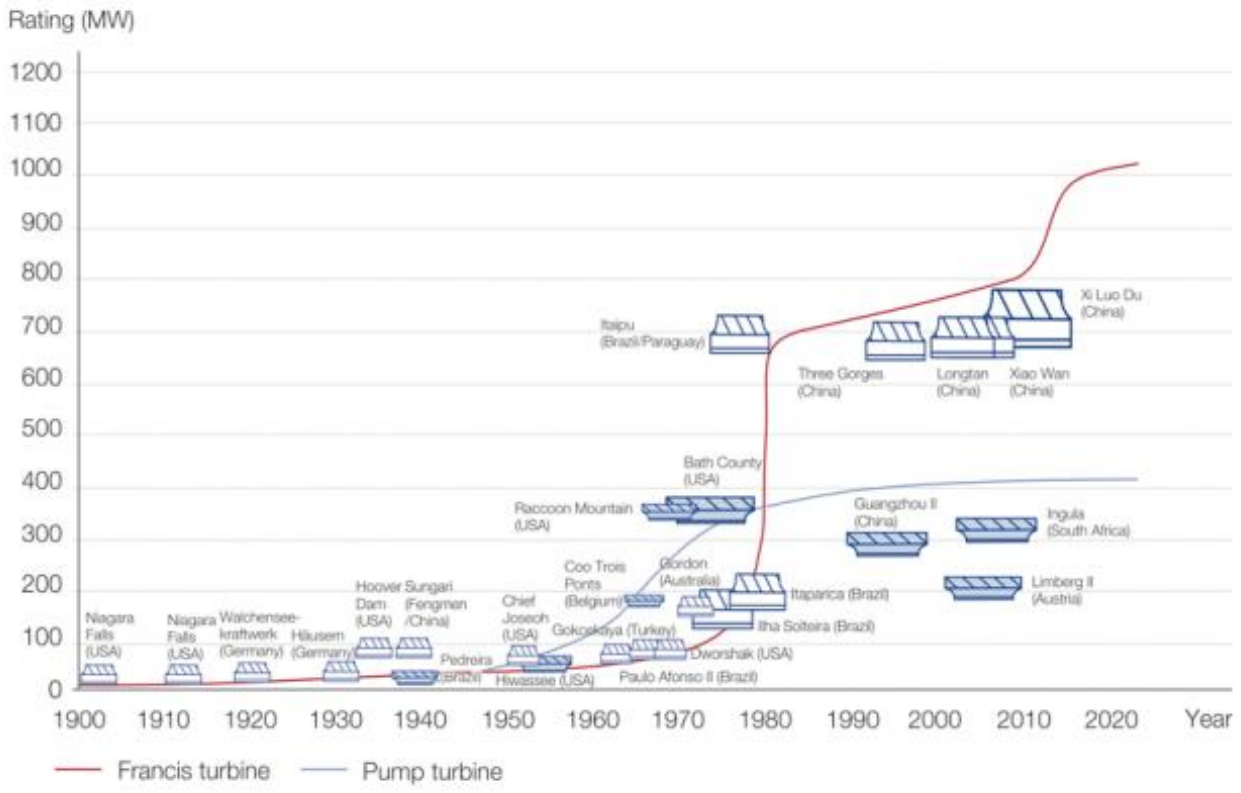


Double-stage pump turbine (H > 700 m)

Image source: Alstom

Francis and Pump- Turbines

Turbine size evolution

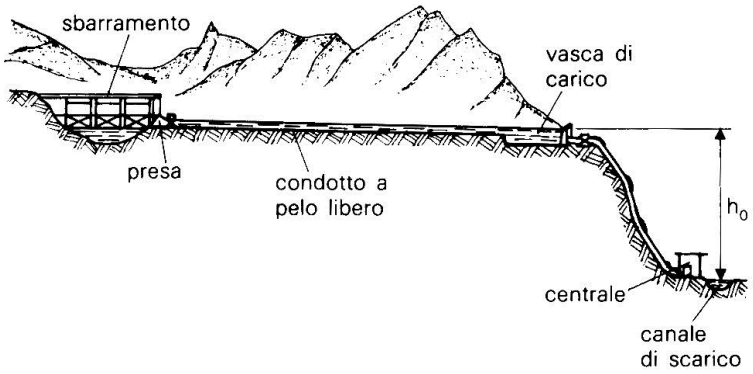


Source: Voith-Siemens

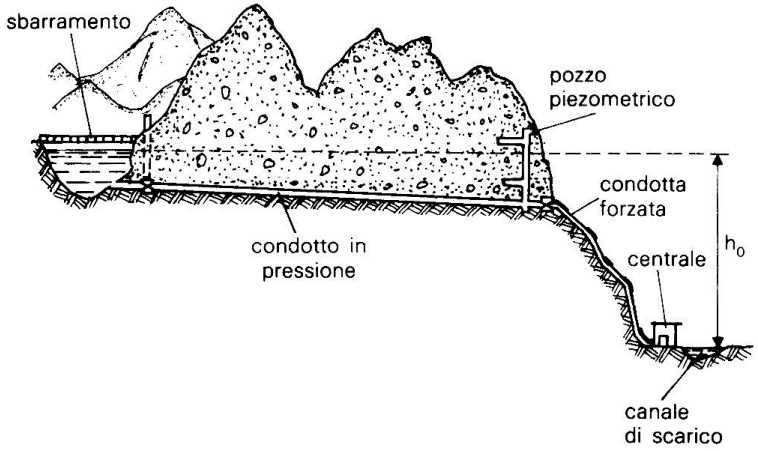
Hydroelectric Power Plants

Classification

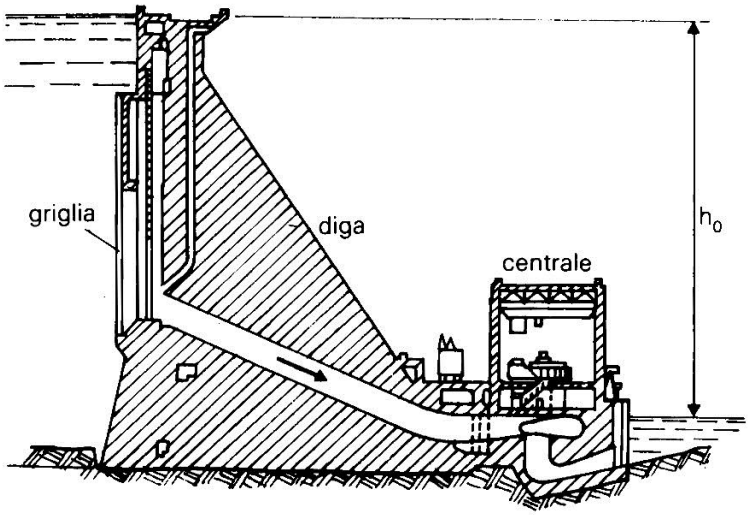
Storage plant:
High head, open channel flow



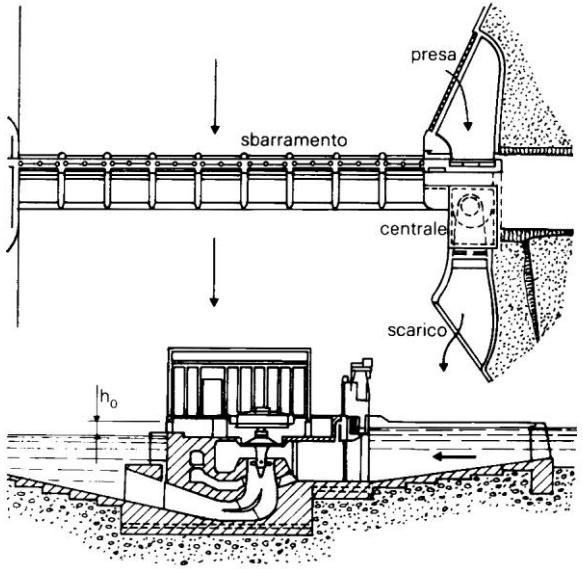
Storage plant:
High head, pipe flow



Storage plant:
Medium head, powerhouse located close to the dam



Run-of-the-river plant (low head)



Hydroelectric Power Plants

Classification

Hydroelectric power plants can be divided in three categories, based on the size of the upstream reservoir: **seasonal storage, weekly or daily storage, run-of-the-river**.

More precisely, the classification is based on the time required in order to supply the reservoir with its nominal capacity, taking the incoming streams at their annual average flow rate (pumped flows excluded).

Hydroelectric power plants are thus classified as follows:

- **seasonal storage reservoirs**: time required to provide nominal capacity > 400 h;
- **weekly or daily storage reservoirs**: time required between 2 and 400 h;
- **run-of-the-river**: plant without upstream reservoir, or whose reservoir needs less than 2 h to reach nominal capacity.

Hydroelectric Power Plants

Pumped storage plants

Pumped storage plants are able to convert electric energy into potential energy by pumping water from a downstream reservoir to an upstream one.

This is economically favorable during so called **off-peak hours**, i.e. when load on the electric grid is low, and a surplus of low-cost electric energy is available, being supplied by base-load power plants.

The energy stored is then converted back into electric energy during peak hours.

The overall system efficiency is usually around 70 ÷ 80%.

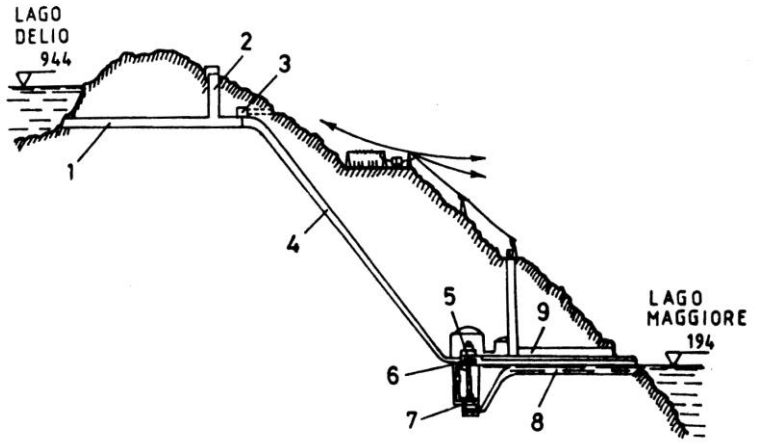


Fig. X.24 - Schema dell'impianto di accumulazione del lago Delio.
1. Opere di presa e galleria di derivazione; 2. Pozzo piezometrico; 3. Camera valvole; 4. Condotta forzata; 5.Motori/Generatori; 6. Turbine Pelton; 7. Pompe multistadio; 8. Canali di scarico; 9. Galleria di accesso.

Image source: R. della Volpe, *Macchine*, Liguori Editore, Napoli, 2011, ISBN:9788820749729.

Hydroelectric Power Plants

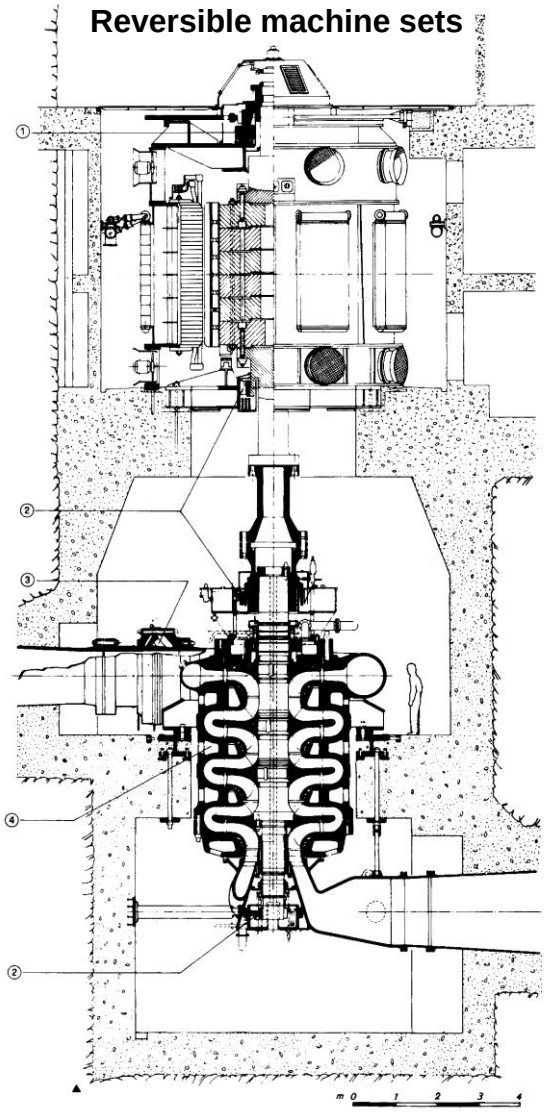
Pumped storage plants

Pumped storage plants can be divided in:

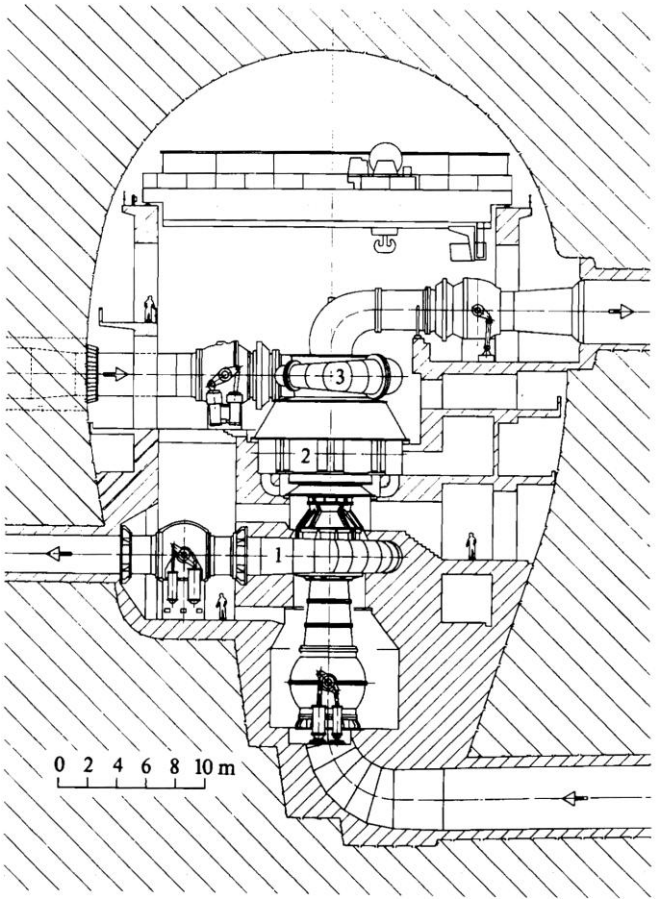
- **ternary systems**: made up of one electric machine and two distinct hydraulic machines (pump and turbine);
- **reversible machine sets**: made up of one electric machine and only one, reversible, hydraulic machine (pump-turbine).

Ternary systems are more suitable for very high heads, with a Pelton turbine and a centrifugal pump.

Reversible machine sets



Ternary system



Hydroelectric Power Plants

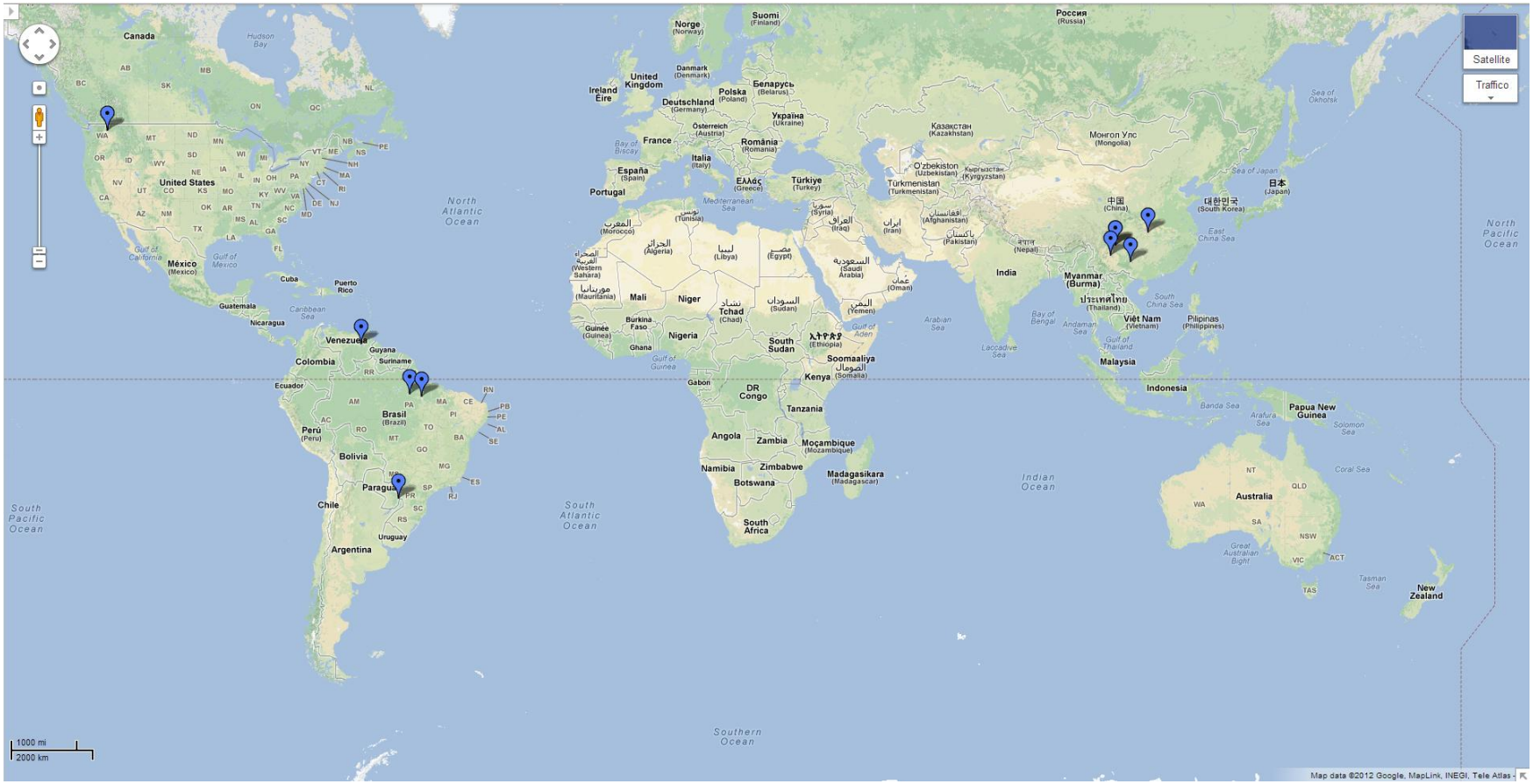
10 largest storage power plants

Plant	Country	Rated power output [GW]	Turbines	Max annual generation [TWh]
Three Gorges Dam	China	22,5	32 x 700 MW Francis 2 x 50 MW Francis	84,4
Itaipu Dam	Brazil/Paraguay	14,0	20 x 700 MW Francis	94,7
Xiluodu Dam*	China	13,9		64,0
Baihetan Dam*	China	13,1		
Belo Monte Dam*	Brasile	11,0	20 x 550÷611 MW Francis 7 x 25,9 MW Kaplan bulb	38,2
Guri Dam	Venezuela	10,2	10 × 730 MW - 4 × 180 MW 3 × 400 MW - 3 × 225 MW 1 × 340 MW	53,4
Wudongde Dam*	China	8,7		
Tucuruí Dam	Brazil	8,4	12 x 350 MW Francis 11 x 375 MW Francis 2 x 22,5 MW (auxiliaries)	41,4
Grand Coulee Dam	USA	6,8	27 Francis 6 pump turbines	20,0
Longtan Dam	China	6,4	9 x 714 MW Francis	18,7

* Under construction

Hydroelectric Power Plants

10 largest storage power plants



Hydroelectric Power Plants

Itaipu power plant (storage plant)

Hydroelectric Power Plants

Itaipu power plant

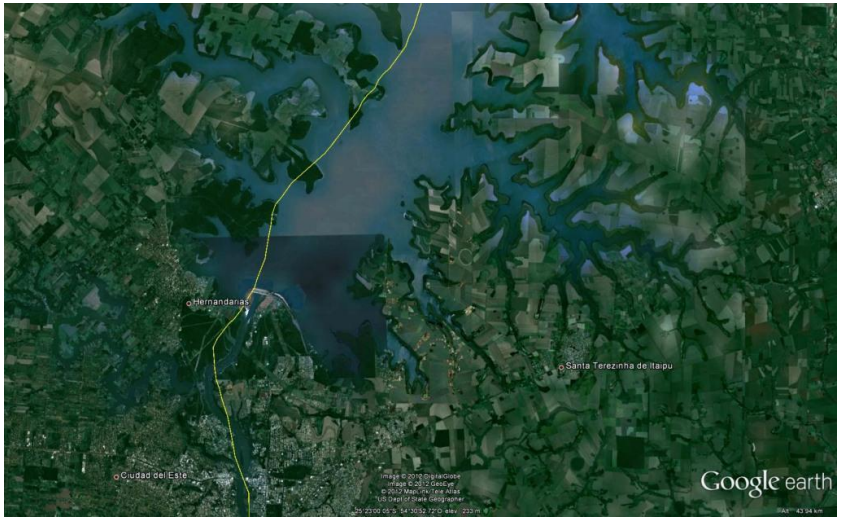
Itaipu power plant (Brazil-Paraguay)

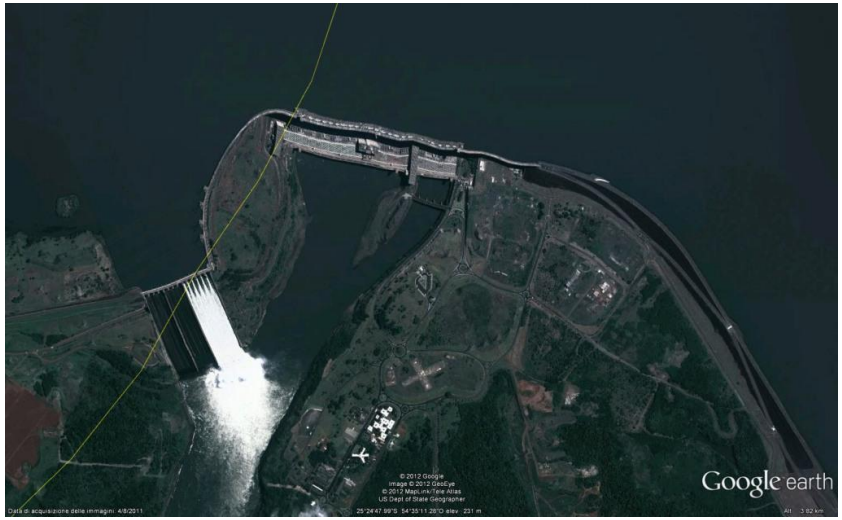
Rated power output	14 GW (= 20 x 700 MW)
Net head	118,4 m
Nominal flow rate	690 m^3/s
Max generation (2008)	94,69 TWh
Turbine type	Francis
Penstocks	10,5 m diameter 142,2 m length

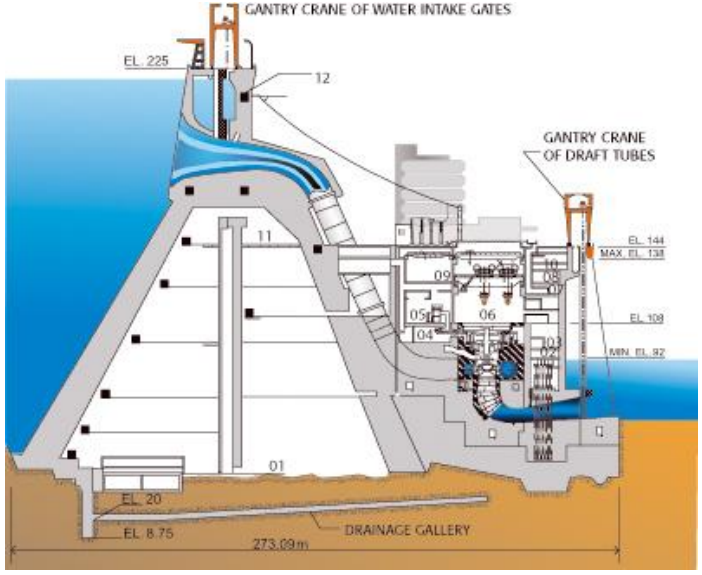
Aerial view © Wikimedia Commons / CC-BY-SA-3.0

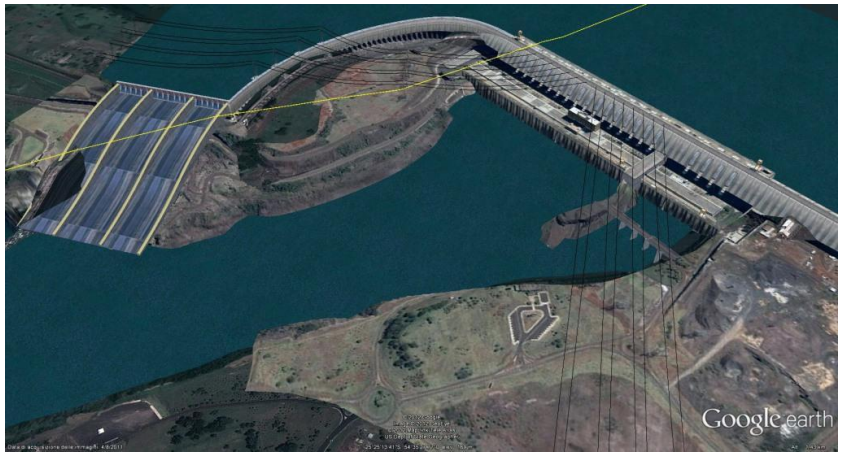
Hydroelectric Power Plants

Itaipu power plant







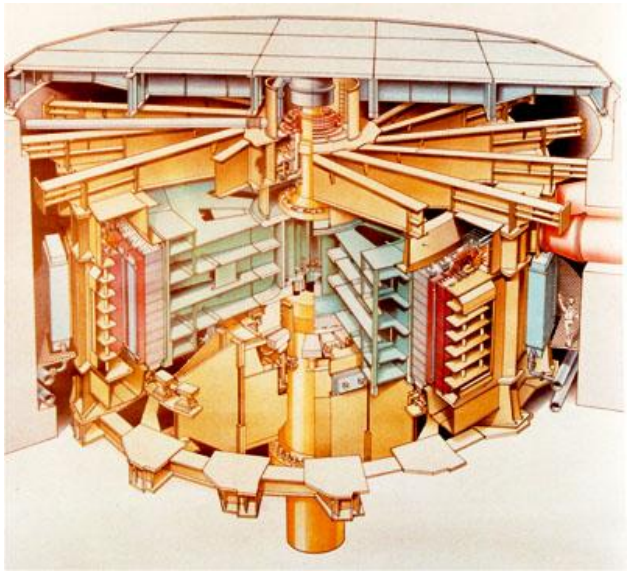


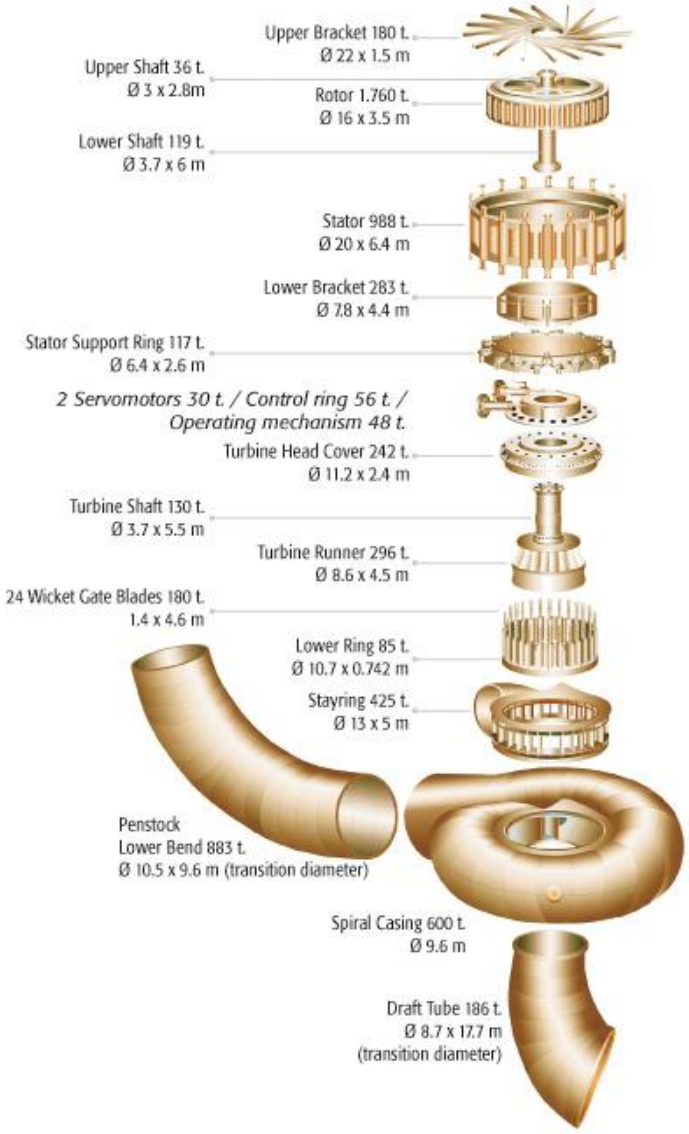
Hydroelectric Power Plants

Itaipu power plant

Electric generators

Quantity	20 units
60 Hz Frequency	10 units
50 Hz Frequency	10 units
50/60 Hz Rated power	823.6/737.0 MVA
Rated voltage	18 kV
Number of 50/60 Hz poles	66/78
Moment of inertia - GD2	320,000 t.m²
50/60 Hz Power factor	0.85/0.95
Heaviest piece - rotor	1,760 t
Weight of each 50/60 Hz unit	3,343/3,242 t



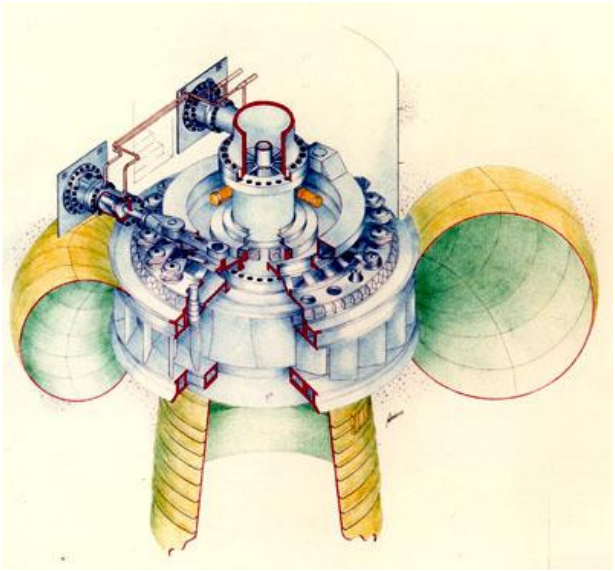

Upper Bracket 180 t.
Ø 22 x 1.5 m
Upper Shaft 36 t.
Ø 3 x 2.8m
Rotor 1.760 t.
Ø 16 x 3.5 m
Lower Shaft 119 t.
Ø 3.7 x 6 m
Stator 988 t.
Ø 20 x 6.4 m
Lower Bracket 283 t.
Ø 7.8 x 4.4 m
Stator Support Ring 117 t.
Ø 6.4 x 2.6 m
2 Servomotors 30 t. / Control ring 56 t. /
Operating mechanism 48 t.
Turbine Head Cover 242 t.
Ø 11.2 x 2.4 m
Turbine Shaft 130 t.
Ø 3.7 x 5.5 m
Turbine Runner 296 t.
Ø 8.6 x 4.5 m
24 Wicket Gate Blades 180 t.
1.4 x 4.6 m
Lower Ring 85 t.
Ø 10.7 x 0.742 m
Stayring 425 t.
Ø 13 x 5 m
Penstock
Lower Bend 883 t.
Ø 10.5 x 9.6 m (transition diameter)
Spiral Casing 600 t.
Ø 9.6 m
Draft Tube 186 t.
Ø 8.7 x 17.7 m
(transition diameter)

Hydroelectric Power Plants

Itaipu power plant

Turbines

Quantity	20
Type	Francis
Unit rated power	715 MW
Project speed - 50/60 Hz	90.9/92.3 rpm
Project net head	118.4 m
Unit rated flow	645 m³/s
Heaviest indivisible piece - rotor	296 t
Weight of each unit	3,360 t



Hydroelectric Power Plants

Run-of-the-river plant: Isola Serafini (PC)

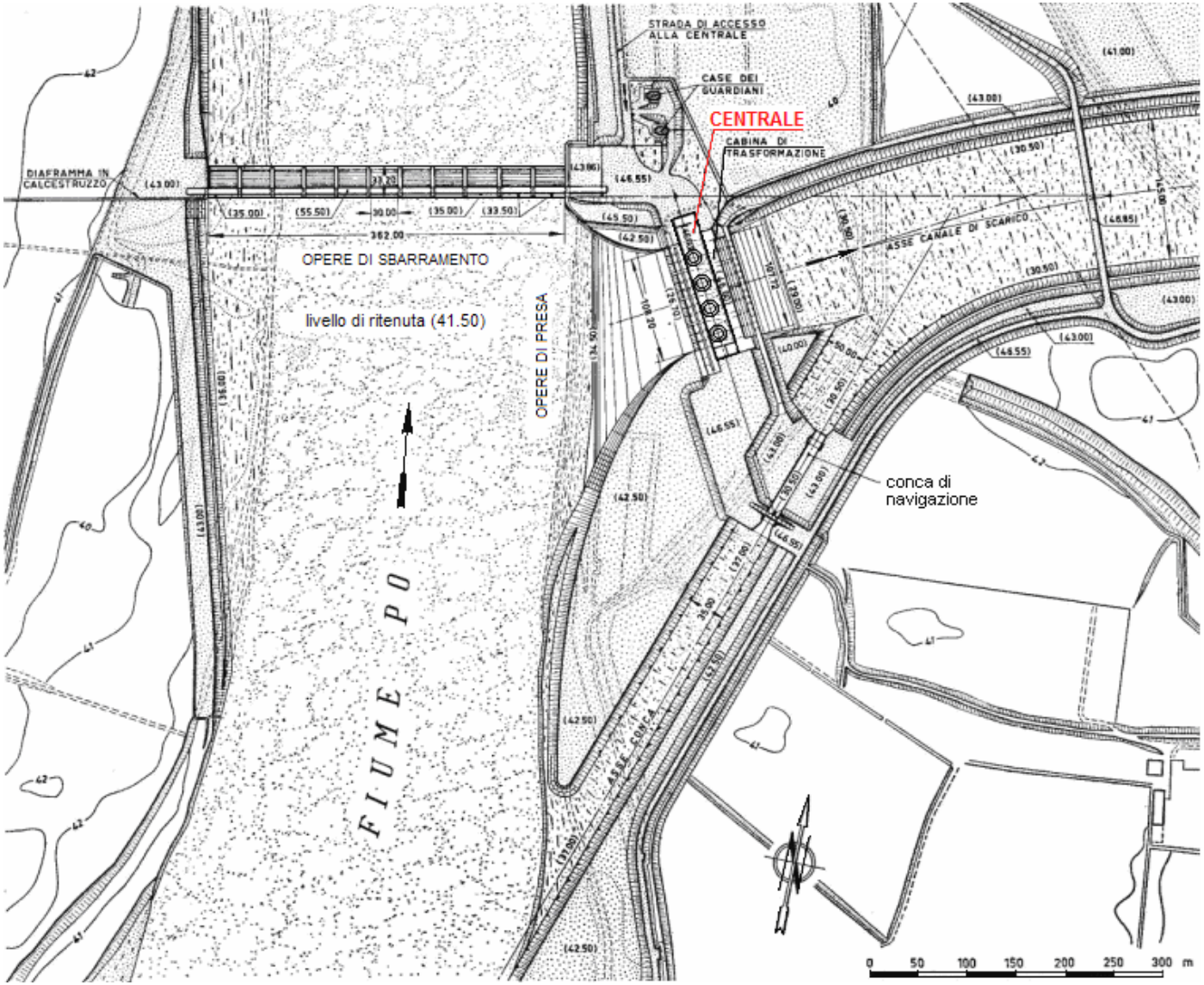
Isola Serafini hydroelectric power plant (PC)

Rated power output	82 MW
Number of turbines	4
Net head (up to)	11 m
Flow rate (up to)	1000 m^3/s
Annual generation*	484 GWh
Type of turbines	Kaplan, vert. axis
Runner diameter	7,6 m
Rotational speed	53,6 rpm
Generator power output	23 MVA
Number of pole pairs	56

* Defined as the maximum electric energy that the plant could produce in a given period if all natural incoming streams are utilized.

Hydroelectric Power Plants

Run-of-the-river plant: Isola Serafini (PC)



Hydroelectric Power Plants

Run-of-the-river plant: Isola Serafini (PC)

Hydroelectric Power Plants

Run-of-the-river plant: Castel Giubileo (RM)

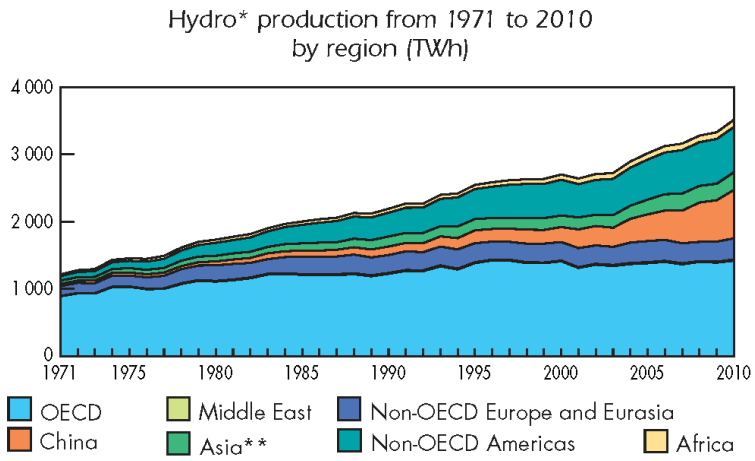
Castel Giubileo power plant (RM)

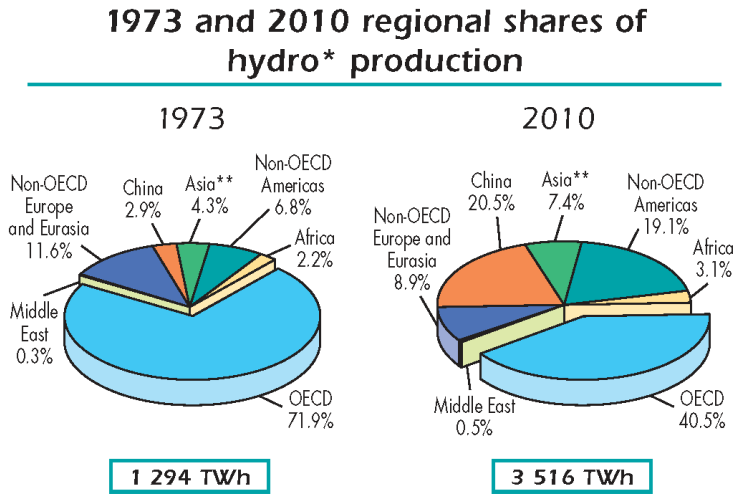
Rated power output	17 MW
Number of turbines	3
Net head	9,58 m
Flow rate	250 m^3/s
Annual generation*	77,09 GWh
Type of turbines	Kaplan, vert. axis

* Defined as the maximum electric energy that the plant could produce in a given period if all natural incoming streams are utilized.

Hydroelectric Power Plants

World hydroelectric generation





Producers	TWh	% of world total
People's Rep. of China	722	20.5
Brazil	403	11.5
Canada	352	10.0
United States	286	8.1
Russian Federation	168	4.8
Norway	118	3.4
India	114	3.3
Japan	91	2.6
Venezuela	77	2.2
France	67	1.9
Rest of the world	1 118	31.7
World	**3 516**	**100.0**

2010 data

*Includes pumped storage.
**Excludes countries with no hydro production.

Installed capacity	GW
People's Rep. of China	171
United States	100
Brazil	79
Canada	75
Japan	47
Russian Federation	47
India	37
Norway	30
France	25
Italy	21
Rest of the world	331
World	**963**

2009 data
Sources: IEA, United Nations.

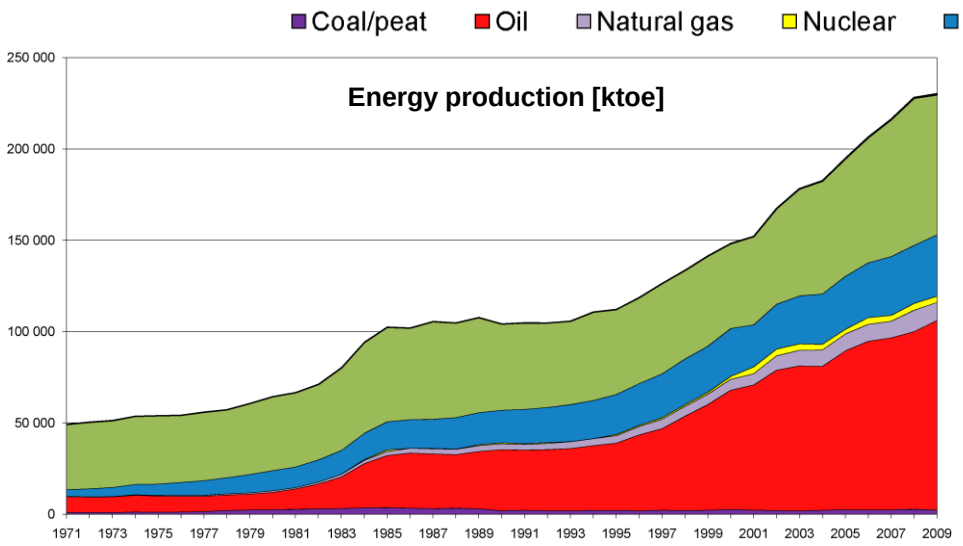
Country (top-ten producers)	% of hydro in total domestic electricity generation
Norway	94.7
Brazil	78.2
Venezuela	64.9
Canada	57.8
People's Rep. of China	17.2
Russian Federation	16.2
India	11.9
France	11.7
Japan	8.1
United States	6.5
Rest of the world**	15.4
World	**16.3**

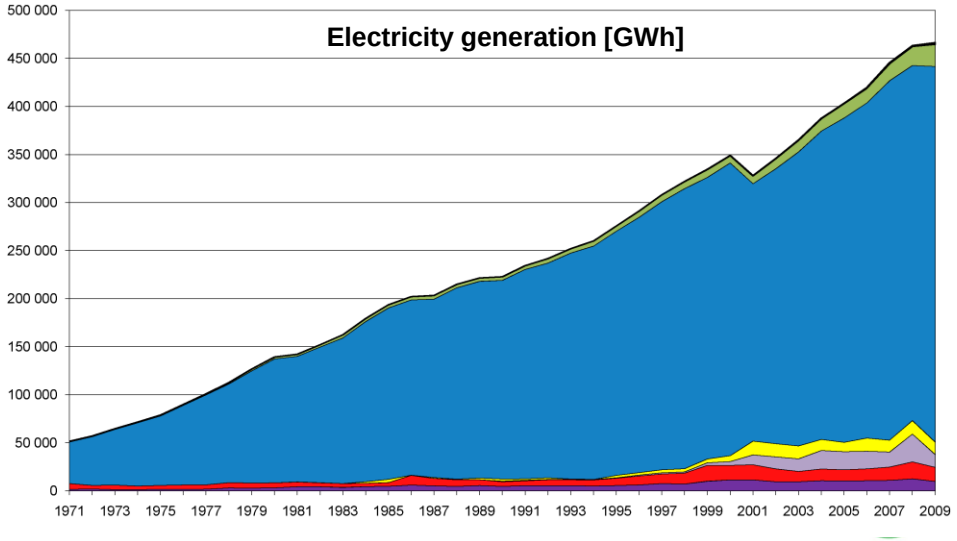
2010 data

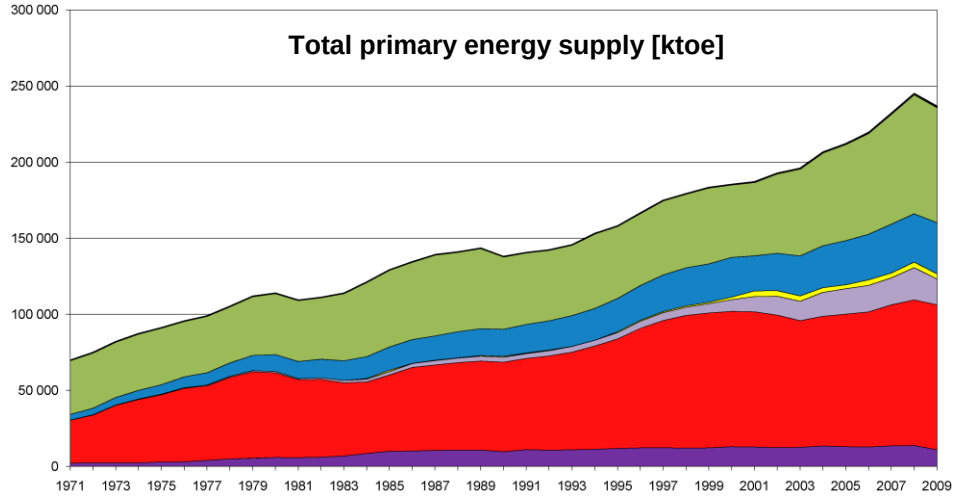
Source: *Key World Energy Statistics 2012*, International Energy Agency

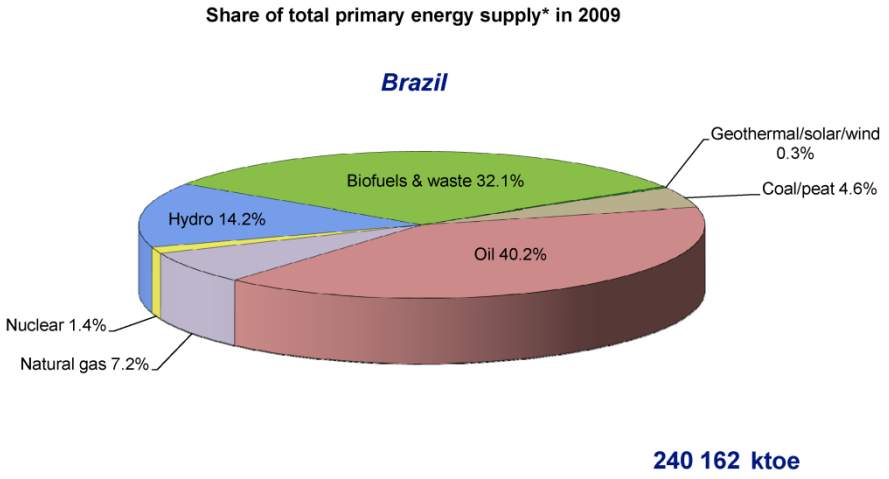
Hydroelectric Power Plants

Brazil's energy generation









Source: International Energy Agency

Hydroelectric Power Plants

Examples of geopolitical issues

South Asia's water Unquenchable thirst

A growing rivalry between India, Pakistan and China over the region's great rivers may be threatening South Asia's peace

Nov 19th 2011

http://www.economist.com/node/21538687

[...] **Half complete, the [Baglihar] dam [...] generates 450 MW for the starved energy grid of Jammu and Kashmir**. Once the scheme fully tames the water, by steering it through a tunnel blasted into the mountain, the grid will gain another 450 MW.

The river swirls away, white-crested and silt-laden, racing to the nearby border with Pakistan. But there Baglihar is a source of bitterness. **Pakistanis cite it as typical of an intensifying Indian threat to their existence**, a conspiracy to divert, withhold or misuse precious water that is rightfully theirs. [...]

More dams are to come, as India's need to power its economy means it is quietly spending billions on hydropower in Kashmir. [...] Some analysts in Srinagar talk of over 60 dam projects, large and small, now on the books.

Any of these could spark a new confrontation. The latest row is over the Kishanganga river [...] as each country races to build a hydropower dam either side of Kashmir's line of control. India's dam will divert some of the river down a 22 km mountain tunnel to turbines. To Pakistani fury, that will lessen the water flow to the downstream dam, so its capacity will fall short of a planned 960 MW. [...]

Hydroelectric Power Plants

Examples of geopolitical issues

Damming the Mekong river River elegy

Laos admits work is going ahead on a controversial dam

Nov 3rd 2012

http://www.economist.com/news/asia/21565676-laos-admits-work-going-ahead-controversial-dam-river-elegy

THE **Mekong river**, snaking its way through the heart of South-East Asia, has long sustained the world's biggest and most productive inland fishery, supplying protein for around 65m mainly poor people from four riparian countries, Laos, Thailand, Cambodia and Vietnam. But scientists warn that **this ecosystem is gravely threatened by the Lao government's rush to exploit its water resources**, egged on by Thai, Chinese and European energy companies.

The decision by Laos to push ahead with the giant Xayaburi dam makes it the first of what could prove to be a cascade of 11 proposed dams on the lower Mekong. Because the decision fails to take account of the consequences for downstream countries, **it has raised tensions with neighbours**. [...]

Many marvels of the Mekong face being wiped out, including the Mekong giant catfish and the Irrawaddy dolphin, as well as the spectacular Khone waterfall. Scientists say the stakes could not be higher. Philip Hirsch at the University of Sydney predicts that the **loss of the fish catch** for millions of Asia's poorest people will prove larger than the entire freshwater catch of Europe and West Africa combined. [...]

Hydroelectric Power Plants

Examples of geopolitical issues

Peru's energy ambitions Hydro-powered dreams

Hopes and fears of a regional energy hub

Feb 10th 2011

http://www.economist.com/node/18114659

AT LESS than 8.000 MW, **Peru**'s total electricity-generation capacity is modest, barely matching four modern nuclear power stations. But President Alan García's government reckons it could produce almost eight times as much power just by **harnessing the country's Amazonian rivers**, [...]

Green groups are mobilising against the proposed hydroelectric dams. Their first target is a $4 billion, 2.000 MW dam at Inambari, in Peru's south-eastern jungle. This would flood around 400 km^2. The protests against it are backed by the regional government. Another dam proposed by a Brazilian consortium, at Paquitzapango, has been stalled by the energy ministry. Leaders of the Asháninkas, an Amazonian tribe, complained that it would displace 10.000 people.

The government's plans centre on the **Marañón river**, which it calls Peru's "energy artery", with the **capacity to generate 10.000 MW from six dams**. But local people along the river say they have not been consulted about the hydroelectric schemes. [...]

Centrali idroelettriche

Aspetti geopolitici

Energy in Brazil Power and the Xingu

A huge Amazon hydropower project shows how hard it is to balance the demands of the environment and of a growing and prospering country

Apr 22nd 2010 http://www.economist.com/node/15954573

[...] **Belo Monte**, a huge hydroelectric power station to be raised on the **Xingu river in the eastern Amazon basin**.

[...] Brazil's rapidly growing economy needs more energy, preferably renewable. The scale of the dam—it will be the **world's third-largest hydroelectric station after China's Three Gorges and Brazil's own Itaipu**—is epic. [...] But ever since the engineers in Brasília rolled out the blueprints for damming the Xingu two decades ago, **the project has attracted powerful opposition**.

Environmental groups and river dwellers say Belo Monte will flood vast patches of rainforest while desiccating others. [...]

A generation ago similar protests over an earlier version of the same dam—known then as Kararao—forced officials to rethink their strategy. They came up with Belo Monte. [...] **Instead of building a great wall across the Xingu to create a massive reservoir, Belo Monte is designed as a run-of-river dam**, [...]

The new version will still flood a lot of forest: a reservoir of 516 km^2 will leave scores of villages awash and force thousands from their homes. But that is a third of the area that the original dam would have inundated. [...]

But **these environmental safeguards will also curb Belo Monte's capacity to generate power**, which will vary with the flow of the Xingu. When swollen by the rainy season, the river will cascade through the turbines, turning out up to 11.200 megawatts—adding 10% to Brazil's existing generating capacity. But during the dry Amazon summer, when the Xingu shrinks, Belo Monte's assured output will plunge to an average of 3,5-4,5 GW. [...]